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U. S. S. *PRUITT*.

DESCRIPTION AND TRIAL PERFORMANCE.

By Charles P. Wetherbee, Associate.

Destroyer No. 347, *Pruitt*, is one of three built by the Bath Iron Works, Ltd., under an order of the President dated July 31, 1918, and confirmed by an order contract dated November 8, 1918. Their numbers are 345, 346 and 347.

These vessels were built on the cost plus fixed fee plan with the builders receiving one-third of any saving in cost below a fixed limit named in the contract.

Bonuses limited to $15,000.00 per vessel for economy in fuel oil consumption below stipulated amounts at various speeds were also provided with equal penalties if these amounts were exceeded.

A speed of 35 knots was guaranteed on a four-hour run when carrying a trial load of 299.1 tons in addition to the completed hull and machinery.

The photograph supplied by Mr. N. L. Stebbins shows the *Pruitt* making 36.11 knots on the measured mile.

LEADING PARTICULARS—HULL—DESIGNED.

Length between perpendiculars, feet and inches	310–0
on L. W. L., feet and inches	310–0
over all, feet and inches	314–4½
Beam, molded, extreme, feet and inches	30–11½
on L. W. L., outside of plating, feet and inches	30–9
Depth at side amidships (molded), feet and inches	20–7¾
Draft of hull to L. W. L., feet and inches	9–3¼
Displacement corresponding, tons	1,200.
Tons per inch at L. W. L.	15.5
Area of immersed midship section, square feet	216.
L. W., plane, square feet	6,500.
Wetted surface, square feet	10,110.
Coefficient of fineness, block	.472
midship section	.754
L. W. L.	.680
Transverse metacentric height, feet	1.82
Center of gravity above base, feet	12.45

BATTERY.

Four 4-inch R. F. Guns.
One 3-inch Anti-Aircraft Gun.
Twelve 21-inch Torpedoes carried in 4 triple tubes.
Two .30 Caliber Machine Guns.

MACHINERY.

Boilers Normand return flame oil-fired on closed fire-room system.

Number	4
Working pressure, pounds per square inch	260
Heating surface, total, square feet	27,000
per boiler, square feet	6,750
Furnace volume, per boiler, cubic feet	718
Burners per boiler	14
Fuel Oil Capacity, U. S. gallons	112,000
Propelling engines, Parsons single reduction geared turbines.	
Propelling engines, total shaft horsepower at 35 knots	27,500
Revolutions per minute at 35 knots	442
Number of shafts	2
Pitch diameter of reduction gear, inches	67.962
Face of reduction gear total, inches	38

Pitch diameter of high pressure pinion, inches	9.916
Ratio of high pressure reduction	6.85
Pitch diameter of low pressure pinion, inches	16.688
Ratio of low pressure reduction	4.07
Number of condensers	2
Cooling surface of each condenser, square feet	7,012
Diameter of shafting, inches	11¼
axial hole in shafting, inches	7¼
propellers, inches	110
Pitch of propellers, inches	122
Projected area of propellers, square feet	36.3
Developed area of propellers, square feet	43.7
Number of blades	3

WEIGHTS AS CONSTRUCTED.

Hull complete, tons	482.39
Machinery complete, including water, tons	428.02
Trial load, tons	299.10
Trial displacement, tons	1,209.51

GENERAL DESCRIPTION OF HULL.

This vessel is of the flush deck type that was used exclusively for the 35-knot class built by the United States Navy during the war. The freeboards at the bow and stern are the same as on the older type of raised forecastle destroyer, thus making a very decided slope of the deck from forward aft.

On the continuous flush main deck are situated forward, a chart house and radio room with a bridge and emergency cabin above them.

Amidships on the main deck is an enclosure having the galley in the center and the forced draft blower inlets at each side.

Aft on the main deck is a house containing an auxiliary radio room, torpedo repair room, a crew's wash room and a crew's water closet.

The interior of the hull between frames 35 to 131 is devoted to the machinery. Forward of the machinery there is a first and a second platform deck. On the first platform deck beginning forward are the following spaces: Lamp room, store room, windlass engine room, quarters for chief petty

officers, ward room and staterooms for the commissioned officers. On the second platform deck are the paint room, chain locker, two store rooms and two compartments devoted to crew's quarters.

The hold in the forward part of the ship contains the peak tanks, a listening room, fuel oil compartments and a 4-inch magazine.

Just abaft the machinery between frames 131 and 137 are fuel oil tanks having the full depth of the vessel and suitably subdivided.

Aft of bulkhead 137 is a single platform deck extending to the stern and containing two crew compartments, a store room and the steering room.

The hold under this deck contains a 4-inch magazine, two store rooms and the after peak tank.

Accommodations for eight commissioned officers and 114 crew are provided.

The boats are as follows:—

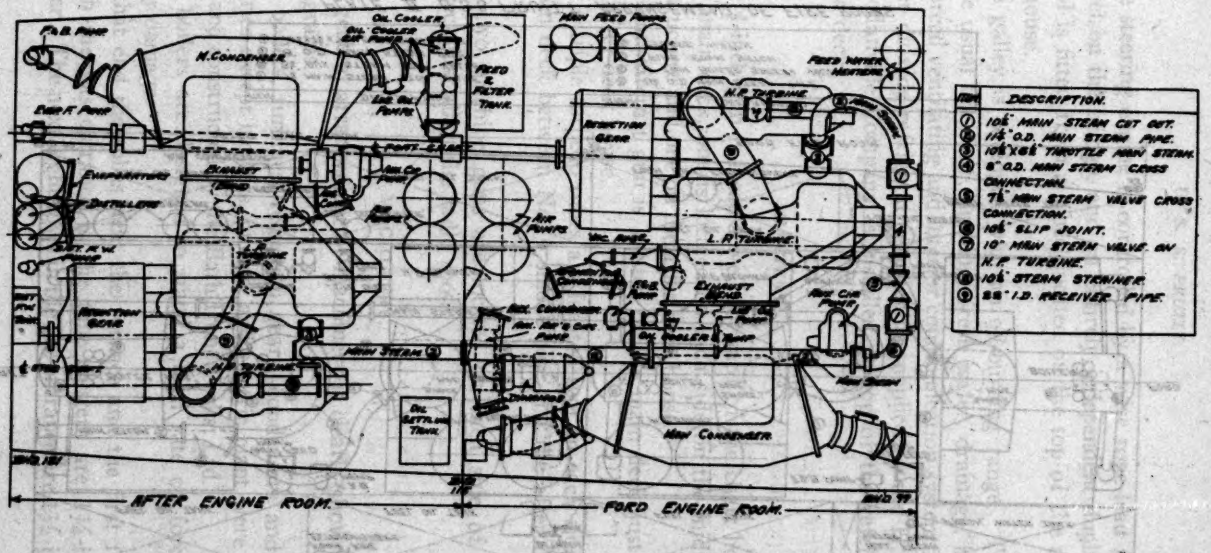
One 24-foot Motor Sailing Launch.
One 24-foot Whaleboat.
One 21-foot Motor Dory.
One 10-foot Punt.

They are all stowed on skids well above the main deck except the whaleboat, which hangs on davits.

HULL AUXILIARIES.

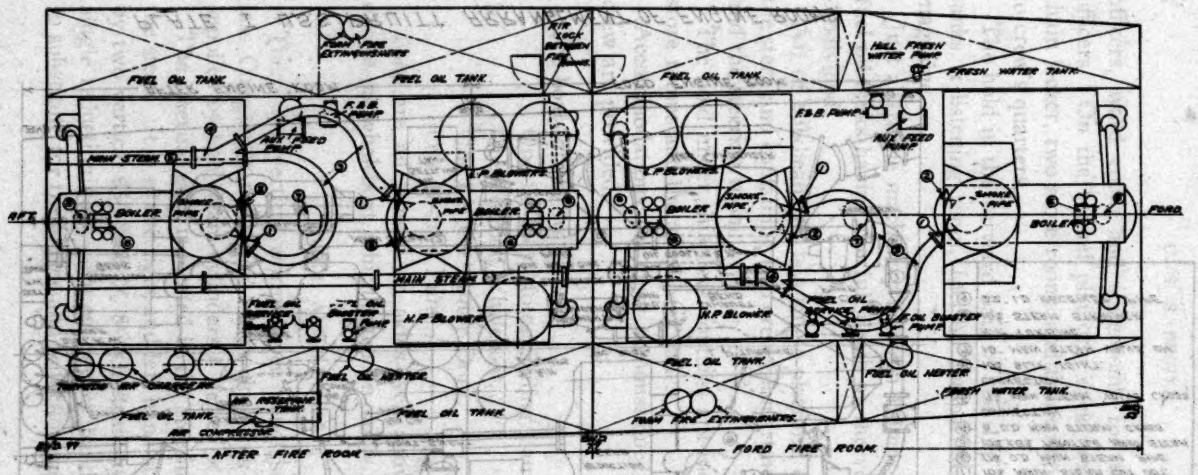
There is a vertical worm-driven windlass of Hyde Windlass Company's make for 1-inch close link chain. Power is supplied by two steam cylinders 5 inches × 4 inches.

The steering gear is located in the steering compartment at the stern. It is of the right and left hand screw type operated by two vertical 6½-inch × 8-inch steam cylinders through a single reduction gear. An emergency hand gear is arranged to substitute for the engine and reduction gear.



ITEM	DESCRIPTION.
1	10½" MAIN STEAM CUT OUT.
2	14" O.D. MAIN STEAM PIPE.
3	10½"x8½" THROTTLE MAIN STEAM.
4	8" O.D. MAIN STEAM CROSS CONNECTION.
5	7½" MAIN STEAM VALVE CROSS CONNECTION.
6	10½" SLIP JOINT.
7	10" MAIN STEAM VALVE ON H.P. TURBINE.
8	10½" STEAM STRAINER.
9	8⅝" I.D. RECEIVER PIPE.

PLATE I U.S.S. PRUITT ARRANGEMENT OF ENGINE ROOMS.



ITEM	DESCRIPTION
2	7½" MAIN STEAM STOP VALVE
3	3½" AUX STEAM STOP VALVE
4	6" O.D. MAIN STEAM PIPE
5	10"X7½"X7½" EXPANSION JOINT

ITEM	DESCRIPTION
6	10" O.D. MAIN STEAM PIPE
7	4" THIN BOILER SAFETY VALVE
8	FIRE ROOM HATCH
9	ESCAPE HATCH

PLATE II U.S.S. PRUITT ARRANGEMENT OF FIRE ROOMS

The steering engine is controlled by a wire rope lead from a wheel on the bridge, and an entirely independent duplicate control is fitted from a wheel located on the top of the after deck house.

The galley is equipped with an oil-burning range. There are the usual fire main, flushing, fresh water, drainage, heating, forced ventilating and interior communication systems. A sulphur dioxide refrigerating machine supplied by the H. W. Johns-Manville Company is installed on top of the refrigerator on deck.

BATTERY.

One 4-inch gun is located well forward of the chart house on the main deck, one is located on each side on the midship superstructure and one on the after deck house.

The anti-aircraft gun is located on the main deck just aft of the after deck house.

The 4 triple torpedo tubes are located two on each side of the main deck between No. 4 smoke pipe and the after deck house.

The two machine guns are located on each side of the main deck just abaft the radio room.

MACHINERY.

The general arrangement of machinery is clearly shown on the diagrams.

The boilers, turbines, reduction gears, condensers and all auxiliaries except blowers, pumps, and electric plant were designed and constructed by the Bath Iron Works, Ltd.

The boilers are designed to stand a high rate of forcing without damage.

To that end the tubes next the furnace and the first four rows in the gas entrance to the tube clusters are 1¾-inch diameter. The next eight rows in the gas entrance are 1½-inch

and the balance are 1¼-inch. The long, small tubes, therefore, come in contact only with comparatively cool gases.

MAIN ENGINES.

Each shaft is driven through a double helical single reduction gear by a high-pressure and a low-pressure turbine of the Parsons type, each operating its own pinion, which meshes with the large gear on the main shaft.

Cruising stages of blading with suitable by-pass belts are fitted on the high-pressure turbines for use at reduced power.

Kingsbury thrust bearings are used to take axial thrust of turbines and also to take the thrust of the propellers.

The condensers are of the bent tube type, with tubes expanded into the tube sheets. The tubes lie in horizontal planes. The circulating water makes a single pass through the tubes.

Circulating water is supplied by scoops without pumps when the vessel is moving ahead. A 14-inch turbine driven, centrifugal pump for each condenser is supplied for use when the vessel is still in the water or going astern.

EVAPORATING PLANT.

There are two evaporators of the Bureau of Engineering's vertical type and two distillers of Bath Iron Works design. The evaporating and distilling plant has a normal capacity of 10,000 gallons of water per 24 hours and an overload capacity of 14,000 gallons. This overload capacity was readily obtained.

LUBRICATING OIL COOLERS.

There are two oil coolers having a total capacity sufficient to cool all the oil required in the forced lubrication service to 100 degrees F. when using cooling water at 60 degrees F.

PUMPS.

The following reciprocating pumps supplied by the Warren Steam Pump Company are installed:

No.	Pumps.	Size (inches)	Kind.	Location.
2	Main air	11 × 32 × 21	Twin, beam, single-acting.	1 in each engine room.
2	Main Feed	16 × 12 × 24	Vertical, piston, double acting, single.	In forward engine room.
2	Auxiliary Feed	15 × 10 × 16	Vertical, piston, double acting, single.	1 in each boiler room.
2	Fire and Bilge.	7 × 7 × 12	Vertical, piston, double-acting, single.	1 in each engine room.
2	Fire and Bilge	7 × 7 × 12	Vertical, piston, double-acting, single.	1 in each boiler room.
1	Auxiliary air and Circ.	6 × 8 × 8 × 7	Horizontal combined.	In forward engine room.
1	Evaporator Feed	4½ × 6 × 6	Vertical, piston, double-acting, single.	In after engine room.
1	Distiller, fresh water.	3½ × 4 × 4	Vertical, piston, double-acting, single.	In after engine room.
3	Lubricating oil	6 × 8 × 12	Vertical, piston, double-acting, single.	1 in forward engine room, 2 in after engine room.
2	Oil cooler circulat'g.	6 × 7 × 8	Vertical, piston, double-acting, single.	1 in each engine room.
4	Fuel-oil service,	6½ × 4½ × 12	Vertical, piston, double-acting, simplex.	2 in each boiler compartment.
2	Fuel-oil booster,	4½ × 6 × 6	Vertical, piston, double-acting, single.	1 in each boiler room.
1	Fresh water.	3½ × 4 × 4	Vertical, piston, double-acting, single.	In forward fire room.

FEED WATER HEATERS.

There are two vertical feed-water heaters of the pressure type capable of condensing the entire exhaust from all auxiliaries at full power.

FORCED-DRAFT BLOWERS.

There are six forced-draft blowers with vertical shafts driven by Terry Turbines. Three are located in each fire room. Any one or any two of these three can operate separately. The turbine of one of them is arranged, however, to operate normally as a high-pressure turbine and exhaust into one or both of the others as low-pressure turbines, thus compounding the steam through the blowers in normal service with a great gain in economy.

FEED AND FILTER TANK.

This tank has a capacity of 1,100 gallons.

AUXILIARY CONDENSER.

There is an auxiliary condenser of 310 square feet cooling surface mounted over a combined horizontal air and circulating pump.

The condenser tubes are bent in a horizontal plane and expanded into the tube sheets.

ELECTRIC PLANT.

Two 25 K.W. direct-connected turbine-driven General Electric Co. generating sets supply direct current at 125 volts for the various services on board.

There are two 24-inch searchlights mounted on a raised platform over the engine room skylight.

The preliminary acceptance trials of this vessel occurred on the Rockland mile and off the Maine Coast August 18 to 21, 1920.

SUMMARY OF DATA

U.S.S. PRUITT [B.I.H - DATA] DATE - AUG. 1

OFFICIAL STANDARDIZATION TRIALS

Δ (5 HIGH RUNS) = 1210 TONS WEATHER [WIND = LIGHT, SEA = SMOOT

NO. OF RUNS			1-2-3		4-5-6		7-8-9		10-11-12		13-14-15		1
SPEED			10.04		15.56		20.98		25.09		27.60		
E.H.P. [Δ = 1210]			230		830		2400		5300		7730		
S.H.P.	PORT		142		784		2085		4484		6609		
	STARBOARD		163		731		1969		4135		6214		
	TOTAL		305		1515		4054		8619		12823		
PROP. EFF'Y.	E.H.P./S.H.P.		.753		.548		.592		.614		.602		
REVOLUTIONS - PORT			106.94		167.01		228.98		287.80		322.63		
	STARBOARD		106.38		167.11		229.36		285.19		320.19		
	MEAN		106.66		167.06		229.17		286.50		321.41		
SLIP (APPARENT)			.0626		.0717		.0888		.123		.144		
EFF. THRUST PER 1" PROJ.			0.713		1.66		3.56		6.57		8.72		
TIP SPEED	FT/MIN		3075		4820		6610		8260		9260		
BOILERS IN USE			4		4		4		4		4		
	ENG. ROOM / FIRE ROOMS		FORD.	AFT	FORD.	AFT	FORD.	AFT	FORD.	AFT	FORD.	AFT	FO
PRESSURES	BOILER STM. GAUGE		247	253	247	250	248	251	248	252	250	250	
	MAIN STM. - E.R. "		247	245	246	244	248	241	248	244	249	241	
	AUX. STM. - E.R. "		242	244	241	242	241	240	246	243	—	—	
	AUX. EXH. - E.R. "		8	7	8	8	6	4	6	7	—	—	
	AUGMENTOR STM. "		160	168	149	150	148	145	149	146	147	143	
	FEED HTR. STM. "		(2) 7	—	(2) 6	—	(2) 5	—	—	—	—	—	
	FUEL OIL TO BURNERS		111	95	105	100	100	100	100	97	101	105	
	LUB. OIL GAUGE BD.		5	6	6	5	6.5	6	7	7	—	—	
TURBINES	H.P. 1ST ROW GAUGE		10	5	50	47	117	113	225	220	239	228	
	H.P. - 3RD EXPAN. "		0	0	18	18	55	52	116	110	181	178	
	H.P. - MAIN BELT "		0	0	0	0	17	14	47	46	76	68	
	H.P. - 7TH EXPAN. "		25"	20"	18"	11"	1	0	17	17	34	29	
	L.P. RECEIVER "		27"	27.5	23"	26"	18"	22"	10.5"	12"	4"	7"	
	VACUUM - (HG) - INS		29.1	—	29.1	—	29.0	—	29.0	—	29.0	—	
MERCURY GAUGE - L.P. TURB.			0.9	—	0.9	—	0.8	—	1.5	—	1.3	—	
(INS. ABSOL.) COND'R TOP			0.9	—	0.9	—	1.0	—	1.0	—	1.0	—	
" BOTT			0.9	—	0.9	—	1.0	—	1.1	—	1.0	—	
AIR PR. SUCT.			2.4	—	2.1	—	1.9	—	2.1	—	2.8	—	
AIR PRESS. F.R. INS.			2.9	3.1	3.0	2.9	3.6	3.4	3.2	3.5	3.5	3.5	
H.P. DUMMY CLEARANCE - INS.			.015	.014	.015	.012	.014	.010	.013	.010	.008	.012	
BOILERS IN USE			2	2	2	2	2	2	2	2	2	2	
BURNERS IN USE			1-1	1-1	2-2	2-2	3-3	3-3	6-6	5-5	7-7	7-7	
BLOWERS IN USE			3	3	3	3	3	3	3	3	3	3	
BLOWER R.P.M.			825	800	825	800	900	850	900	870	925	900	
F.O. SERVICE PP. R.P.M.			2	2	3	3	4	5	9	6	9	9	
TEMP. °F	AIR PUMP DISCH.		84	—	70	—	68	—	67	—	68	—	
	MAIN INJECTION		58	58	58	57	56	53	56	55	58	56	
	MAIN DISCHARGE		62	64	60	62	60	60	61	61	63	63	
	GASES IN UPTAKE		380	—	400	—	425	—	490	—	580	—	
	FIRE ROOMS		104	109	104	110	104	104	104	103	105	104	
	F.O. TO HEATER		79	79	79	79	78	76	76	76	76	76	
	F.O. FROM HEATER		200	202	168	160	164	154	164	155	165	168	
WATER - LBS. PER HR. - A.P.			——		——		——		——		——		
LBS./S.H.P./HR. - A.P.			——		——		——		——		——		
AUX. EXH. (SURPLUS) LED TO			M. COND'R		M. COND'R		M. COND'R		M. COND'R		M. COND'R		M

ATE - AUG. 18-1920

ND = LIGHT - N.E.
A = SMOOTH

13-14-15		16-17-18		19-20-21		22 TO 26		BACKING	
2760		30.08		32.91		35.85		20.79	
7730		10310		13120		16100		—	
6609		8723		11374		14850		3015	
6814		8474		11703		14660		3023	
18823		17197		23077		29610		6048	
.602		.600		.568		.546		—	
322.63		355.61		401.59		459.25		271.06	
320.19		356.37		402.67		460.05		272.19	
321.41		355.99		402.13		459.65		271.63	
.144		.157		.183		.223		.238	
8.78		10.67		12.42		13.98		—	
9260		10350		11600		13260		7820	
4		4		4		4		4	
FORD	AFT	FORD	AFT	FORD	AFT	FORD	AFT	FORD	AFT
250	250	248	252	248	254	252	255	—	—
242	241	246	248	241	240	242	238	260	247
—	—	—	—	240	—	239	240	—	—
—	—	—	—	6	—	8	12	—	—
147	143	145	142	144	145	139	149	—	—
—	—	—	—	(2)5	—	(2)4	—	—	—
101	105	100	103	103	98	118	119	—	—
—	—	—	—	—	—	7.5	7.5	—	—
239	228	233	226	222	226	219	220	143	148
181	178	187	193	201	178	222	221	—	—
75	68	113	112	164	165	219	218	—	—
34	39	53	63	78	83	110	111	—	—
4"	7"	3	3	9	10	16	16.3	—	—
29.0	—	28.8	—	28.5	—	27.9	—	26.4	—
1.3	—	1.8	—	2.2	—	2.7	—	3.3	—
1.0	—	1.2	—	1.5	—	2.1	—	2.4	—
1.0	—	1.1	—	1.3	—	1.2	—	3.3	—
2.8	—	2.4	—	2.7	—	2.4	—	4.1	—
3.5	3.5	3.5	3.5	4.2	4.1	4.6	4.5	—	—
.018	.018	.020	.017	.027	.023	.027	.027	.022	.019
2	2	2	2	2	2	2	2	—	—
7-7	7-7	9-9	9-9	13-13	12-12	14-14	14-14	—	—
3	3	3	3	3	3	3	3	—	—
925	900	960	945	1080	1050	1115	1115	—	—
9	9	13	12	18	14	21	20	—	—
68	—	69	—	70	—	74	—	—	—
58	56	58	56	58	56	58	57	61	61
63	63	65	67	67	69	70	71	104	104
520	—	540	—	600	—	620	—	—	—
105	104	106	100	103	97	113	100	—	—
76	76	76	76	76	76	76	76	—	—
165	168	170	175	173	174	172	169	—	—
——		——		——		——		235608	
——		——		——		——		40.2	
M. COND'R		M. COND'R		M. COND'R		M. COND'R		M. COND'R	

SUMMARY OF DATA

U.S.S. PRUITT — [B.I.W. - DATA] — DATE-AUG. 19-21, 1920

OFFICIAL TRIALS

TRIAL		4 HOURS AT 15 KNOTS		4 HOURS AT 20 KNOTS		6 HOURS AT 25 KNOTS		4 HOURS AT 30 KNOTS		4 HOURS AT FULL SPEED	
DATE		AUG. 20-1920		AUG. 19-1920		AUG. 21-1920		AUG. 20-1920		AUG. 19-1920	
DRAFT		FORD.	AFT	FORD.	AFT	FORD.	AFT	FORD.	AFT	FORD.	AFT
	BEGINNING	9'-9 1/4"	9'-0"	9'-4 1/8"	9'-6 1/8"	9'-1 1/2"	9'-8 5/8"	9'-2 1/2"	9'-8 1/2"	9'-6 5/8"	9'-6 1/8"
	END	9'-8 5/8"	9'-5/8"	9'-4"	9'-4 1/8"	9'-2 3/4"	9'-4 1/2"	9'-5"	9'-2 1/2"	9'-3 3/4"	9'-3 1/4"
	MEAN	9'-4 5/8"		9'-4 1/2"		9'-4 1/4"		9'-4 5/8"		9'-5"	
DISPLACEMENT	MIDDLE OF TRIAL	1212		1210		1213		1215		1217	
SPEED	AVERAGE	15.03		20.25		25.02		30.07		35.09	
S.H.P.	PORT	681		1816		4363		8656		14000	
	STARBOARD	692		1810		4200		8594		13862	
	TOTAL	1373		3626		8563		17250		27862	
REVOLUTIONS - PORT		161.13		220.60		284.80		355.99		443.72	
	STARBOARD	161.14		219.59		285.92		356.10		444.38	
	MEAN	161.14		220.10		285.36		356.05		444.05	
WATER MADE BY EVAPS GALS		—		396.3 3 HRS. - D.EFR.		—		—		—	
BOILERS IN USE		1		2		2		4		4	
WEATHER - WIND		LIGHT-S.W.		LIGHT-S.W.		LIGHT-S		LIGHT AIRS		LIGHT AIRS	
SEA		SMOOTH		SMOOTH		SMOOTH		SMOOTH		SMOOTH	
ENG. RM FIRE ROOMS		FORD.	AFT	FORD.	AFT	FORD.	AFT	FORD.	AFT	FORD.	AFT
BOILER	GUAGE	—	201	—	204	—	254	256	254	258	258
MAIN STM. - E.R.	"	195	192	196	194	248	241	247	241	238	236
AUX. STM. - E.R.	"	192	191	193	192	243	240	245	234	242	238
AUX. EXH. - ER.	"	11	9	7	8	8	10	8	7	7	8
AUGMENTOR STM.	"	71	76	67	75	92	95	100	101	118	127
FEED HT B. STM.	"	(1) 10	—	(1) 6	—	(2) 9	—	(2) 5	—	(2) 4	—

PRESSURES	AUX. STM. - E.R. "	192	191	193	192	243	240	245	234	242	238
	AUX. EXH. - ER. "	11	9	7	8	8	10	8	7	7	8
	AUGMENTOR STM. "	71	76	67	75	92	95	100	101	118	127
	FEED HT & STM. "	(1)10	—	(1)6	—	(2)9	—	(2)5	—	(2)4	—
	TRAP DRAIN MAIN "	3.5"	1#	0	2"	2#	3#	3#	2#	12"	11"
	FUEL OIL TO BURNERS	—	90	—	99	—	153	110	85	135	140
	LUB. OIL - GAUGE BD.	4	5	6	6	6.5	5.5	7	7	7	8
	TURBINES H.P. - 1ST. ROW GAUGE	49	39	106	112	232	229	227	224	221	221
	H.P. - 3RD EXPAN. "	17	16	48	51	119	113	226	222	210	207
	H.P. - MAIN BELT "	0	0	16	10	48	39	109	102	205	204
	H.P. - 7TH EXPAN. "	18"	10"	3"	1.4#	17	17	52	49	103	104
	L.P. RECEIVER "	24.5"	25.5"	20.5"	21"	13.5"	13.5"	2#	1#	14#	14#
	VACUUM - (HG) INS.	28.6	—	28.1	—	28.6	—	28.5	—	28.2	—
	MERCURY GAUGE - L.P. TURB.	1.5	—	2.0	—	1.4	—	1.8	—	2.6	—
	(INS. ABSOL.) COND & TOP	1.4	—	1.9	—	1.4	—	1.5	—	1.8	—
	" BOTT.	1.4	—	1.6	—	1.4	—	1.4	—	1.3	—
	AIR PP. SUCT.	2.4	—	2.7	—	2.3	—	2.5	—	2.8	—
	AIR PRESS. - F.R. INS.	—	1.0	—	1.2	—	2.4	2.0	2.0	4.4	4.4
REVS.	MAIN FEED PUMPS	(1)5	—	(1)6	—	(2)6	—	(2)11	—	(2)16	—
	MAIN AIR "	—	10	—	13	12	13	9	13	12	12
	OIL COOLER CIRC. "	29	21	46	16	47	56	61	58	68	74
	LUB. OIL	16	17	23	23	28	24	31	28	35	40
H.P. DUMMY CLEARANCE-INS.		.015	.013	.018	.010	.013	.014	.023	.019	.027	.027
BOILERS IN USE		—	1	—	2	—	2	2	2	2	2
BURNERS IN USE		—	5	—	5-5	—	8-7	9-9	10-9	14-14	14-14
BLOWERS IN USE		—	2	—	2	—	3	3	3	3	3
BLOWER R.P.M.		—	487	—	564	—	770	735	727	1043	1043

FOLD HERE

TURES - OF	FEED HEATER DISCH.	(1)230	—	(1)208	—	(2)219	—	(2)189	—	(2)188	—
	AIR PUMP DISCH.	69	—	73	—	70	—	71	—	77	—
	L. OIL TO COOLER	80	79	85	88	92	92	100	100	113	111
	L. OIL FROM COOLER	73	74	78	82	83	84	89	88	99	99
	CIRC. WAT. FROM COOLER	62	63	65	68	64	66	63	64	68	69
	MAIN INJECTION	58	58	61	60	61	62	58	58	62	62

PRESSURES	AUGMENTOR STM. "											
	FEED HT & STM. "	(1) 10	—	(1) 6	—	(2) 9	—	(2) 5	—	(2) 4	—
	TRAP DRAIN MAIN "	3.5"	1#	0	2"	2#	3#	3#	2#	12"	11"
	FUEL OIL TO BURNERS	—	90	—	99	—	153	110	85	135	140
	LUB. OIL - GAUGE BD.	4	5	6	6	6.5	5.5	7	7	7	8
	TURBINES H.P. - 1ST ROW GAUGE	49	39	106	112	232	229	227	224	221	221
	H.P. - 3RD EXPAN. "	17	16	48	51	119	113	226	222	210	207
	H.P. - MAIN BELT "	0	0	16	10	48	39	109	102	205	204
	H.P. - 7TH EXPAN. "	18"	10"	3"	1.4#	17	17	52	49	103	104
	L.P. RECEIVER "	24.5"	25.5"	20.5"	21"	13.5"	13.5"	2#	1#	14#	14#
	VACUUM - (HG) INS.	28.6	—	28.1	—	28.6	—	28.5	—	28.2	—
	MERCURY GAUGE-L.P. TURB.	1.5	—	2.0	—	1.4	—	1.8	—	2.6	—
	(INS. ABSOL.) COND & TOP	1.4	—	1.9	—	1.4	—	1.5	—	1.8	—
	" BOTT.	1.4	—	1.6	—	1.4	—	1.4	—	1.3	—
	AIR PP. SUCT.	2.4	—	2.7	—	2.3	—	2.5	—	2.8	—
	AIR PRESS. - F.R. INS.	—	1.0	—	1.2	—	2.4	2.0	2.0	4.4	4.4
REVS.	MAIN FEED PUMPS	(1) 5	—	(1) 6	—	(2) 6	—	(2) 11	—	(2) 16	—
	MAIN AIR "	—	10	—	13	12	13	9	13	12	12
	OIL COOLER CIRC. "	29	21	46	16	47	56	61	58	68	74
	LUB. OIL	16	17	23	23	28	24	31	28	35	40
H.P. DUMMY CLEARANCE-INS.		.015	.013	.018	.010	.013	.014	.023	.019	.027	.027
BOILERS IN USE		—	1	—	2	—	2	2	2	2	2
BURNERS IN USE		—	5	—	5-5	—	8-7	9-9	10-9	14-14	14-14
BLOWERS IN USE		—	2	—	2	—	3	3	3	3	3
BLOWER R.P.M.		—	487	—	564	—	770	735	727	1093	1093

FOLD HERE

TEMPERATURES - °F	FEED HEATER DISCH.	(1) 230	—	(1) 208	—	(2) 219	—	(2) 189	—	(2) 188	—
	AIR PUMP DISCH.	69	—	73	—	70	—	71	—	77	—
	L. OIL TO COOLER	80	79	85	88	92	92	100	100	113	111
	L. OIL FROM COOLER	73	74	78	82	83	84	89	88	99	99
	CIRC. WAT. FROM COOLER	62	63	65	68	64	66	63	64	68	69
	MAIN INJECTION	58	58	61	60	61	62	58	58	62	62
	MAIN DISCHARGE	59	60	64	62	72	70	69	69	74	78
	F. OIL TO HEATER	—	83	—	78	—	74	75	73	77	76
	F. OIL FROM HEATER	—	178	—	171	—	196	171	174	170	167
	GASES IN UPTAKE	—	—	—	—	—	—	527	—	618	—
	ENGINE ROOMS	91	84	90	87	85	85	84	84	89	—
	FIRE ROOMS	—	119	—	129	—	114	120	120	—	—
	LBS./HP.-hr. SHAFT H.P.	32144		58073		111942		210363		346315	

TEMP.	GASES IN UPTAKE	—	—	—	—	—	—	527	—	618	—
	ENGINE ROOMS	91	84	90	87	85	85	84	84	89	—
	FIRE ROOMS	—	119	—	129	—	114	120	120	—	—
WATER	LBS./HR - INCL. EVAPS - A.P.	32144		59072		111942		210363		345315	
	" " - NO AUX EX - T.O	24751		52243		95755		183975		307875	
	LBS./S.H.P./HR. - A.P.	23.4		16.3		13.1		12.2		12.4	
	" " " - T.O.	17.3		14.4		11.2		11.65		11.0	
	EVAP PER LB. OIL - ACT.	15.8		15.1		15.1		14.8		14.4	
	" " " " F&A. 212°	16.3		15.9		15.8		15.95		15.5	
	LBS. PER ℔ H.S. - A.P.	4.7%		4.37		8.3		7.8		12.8	
FUEL OIL	CALORIFIC VAL. - B.T.U./LB	19124		19053		19124		19124		19053	
	SPECIFIC GRAV. - 60°F	0.9266		0.9261		0.9266		0.9266		0.9264	
	LBS. PER HOUR - LAST 3 HRS MEASURED	2034		3918		7390 LAST 5 HRS.		14195		23999	
	LBS./S.H.P./HR. - LAST 3 HRS MEASURED	1.48		1.08		0.863 LAST 5 HRS.		0.823		0.861	
	LBS./℔ H.S. - LAST 3 HRS MEASURED	0.310		0.290		0.548 LAST 5 HRS.		0.526		0.889	
	LBS. PER KNOT - LAST 3 HRS MEASURED	135.2		193.1		295.0 LAST 5 HRS.		472.0		684.0	
	LBS. PER KNOT - LAST 3 HRS. GOVT. CORR'D TO 19500 B.T.U. AND EVEN SPEED	133.5		185.5		289.0 LAST 5 HRS		459.5		663.0	
	LBS. PER KNOT - GUARANTEE CORR'D TO 19500 B.T.U. AND EVEN SPEED	155		202		348		558		772	
	CR. RADIUS ON 375 TONS	6290		4530		2905		1828		1267	
AUX. EXH. LED TO - 1ST HR. (SURPLUS) REMAINDER		AUX. COND. M. BELT - AFT 3/4 TURNS		AUX. COND. ——		AUX. COND. 7TH EXP. AFT 2 TURNS		AUX. COND. ——		AUX. COND. L.A RECEIVER BOTH E.R.	
TRAP DRAIN LED TO 1ST HR. REMAINDER		AUX. COND. MAIN. "		AUX. COND. M. "		AUX. COND. M. "		AUX. COND. M. "		AUX. COND. M. "	

NOTE:- DURING 1ST HOUR OF SEA TRIAL CONDENSATE FROM STEAM TO TURBINES ONLY WAS MEASURED.

DURING REMAINDER OF SEA TRIAL CONDENSATE FROM STEAM FOR ALL PURPOSES WAS MEASURED.

The contract provides for—

(1) A standardization trial on the measured mile to consist of a total of 26 runs at speeds from 10 knots to maximum speed, and also two runs astern at 20 knots.

(2) A four-hour run at full power.

(3) A four-hour run at 30 knots.

(4) A six-hour run at 25 knots.

(5) A four-hour run at 20 knots.

(6) A four-hour run at 15 knots.

(7) Turning trials both ahead and astern, stopping, backing and starting trials.

These trials were all carried out without incident. The results observed are shown on the tabulated sheet and the sheet of curves attached.

On trials (2) to (6), inclusive, fuel oil consumption was measured throughout the trials. During the first hour of each of these trials the water consumption measured was that of turbines only, the water consumption recorded for the rest of each run was for all purposes.

The difference between the consumption during the first hour of the run and the hourly consumption during the subsequent hours gave the steam consumption of the auxiliaries.

On trial (5) the evaporators and distillers were run at a rate of 2,620 gallons of water per 24 hours, and, of course, this increased the consumption by the auxiliaries on this trial.

The vessel was backed steadily and continuously at a speed of 20.7 knots for a period of 25 minutes during the two runs astern on the measured mile, and while conducting the steering tests going astern. This gave an opportunity to measure the water consumption for all purposes and the shaft horsepower while going astern at a speed of 20.7 knots. The result is given in the table.

The sheet of curves permits some interesting observations.

The curves of oil consumption per shaft horsepower and water consumption per shaft horsepower both for all purposes and for turbines only are very flat throughout a very wide range. The oil consumption and water consumption per horsepower are practically constant for all powers from 25 per cent to 100 per cent of full power, and the loss of economy below 25 per cent of full power is not at all rapid. This flat characteristic of these curves is in marked contrast to that of similar curves for direct coupled turbine drives. The reduction gear permits the use of a turbine that makes this improvement possible.

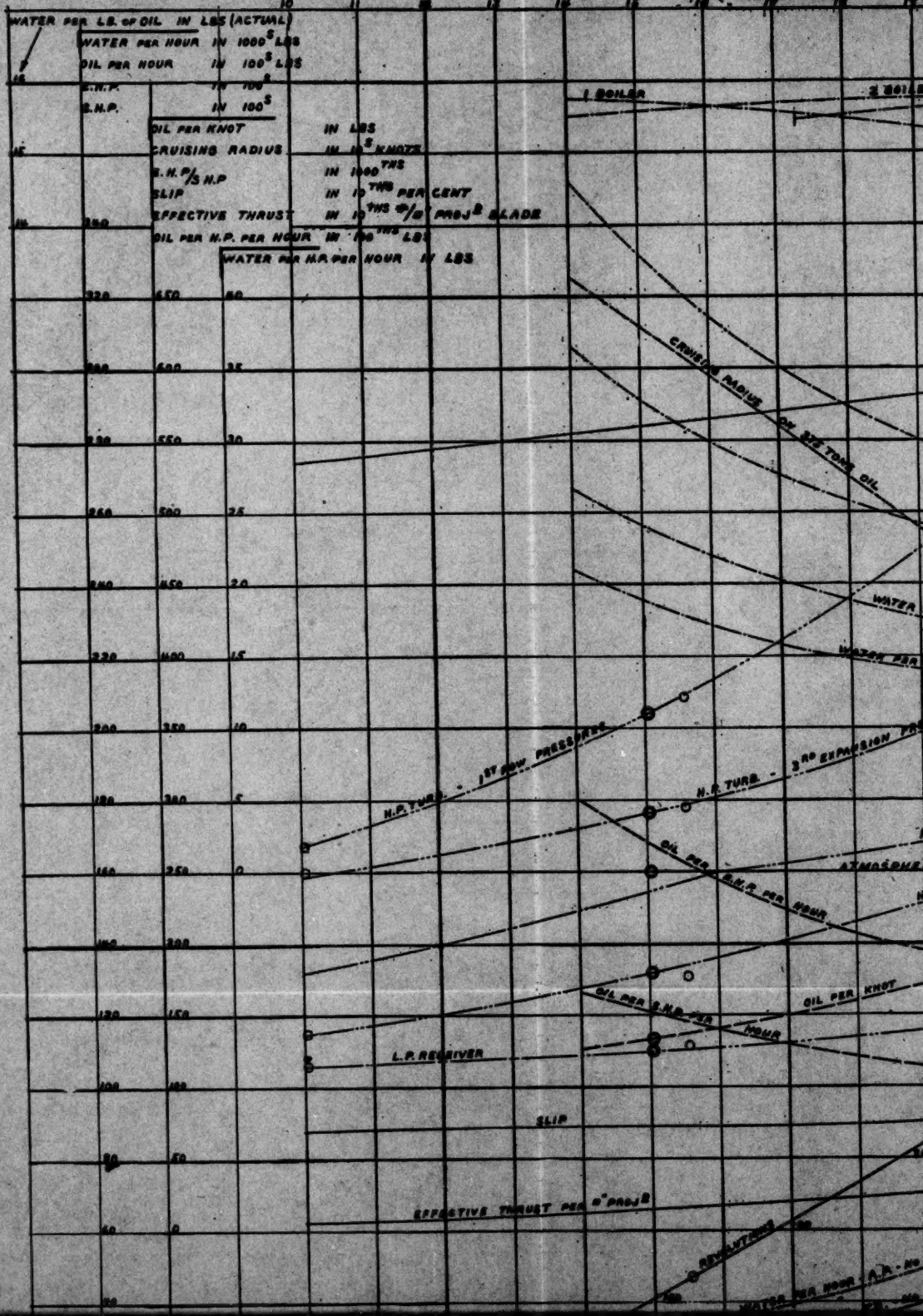
The best fuel and water economies occur in the neighborhood of 29 knots and they are as follows:—

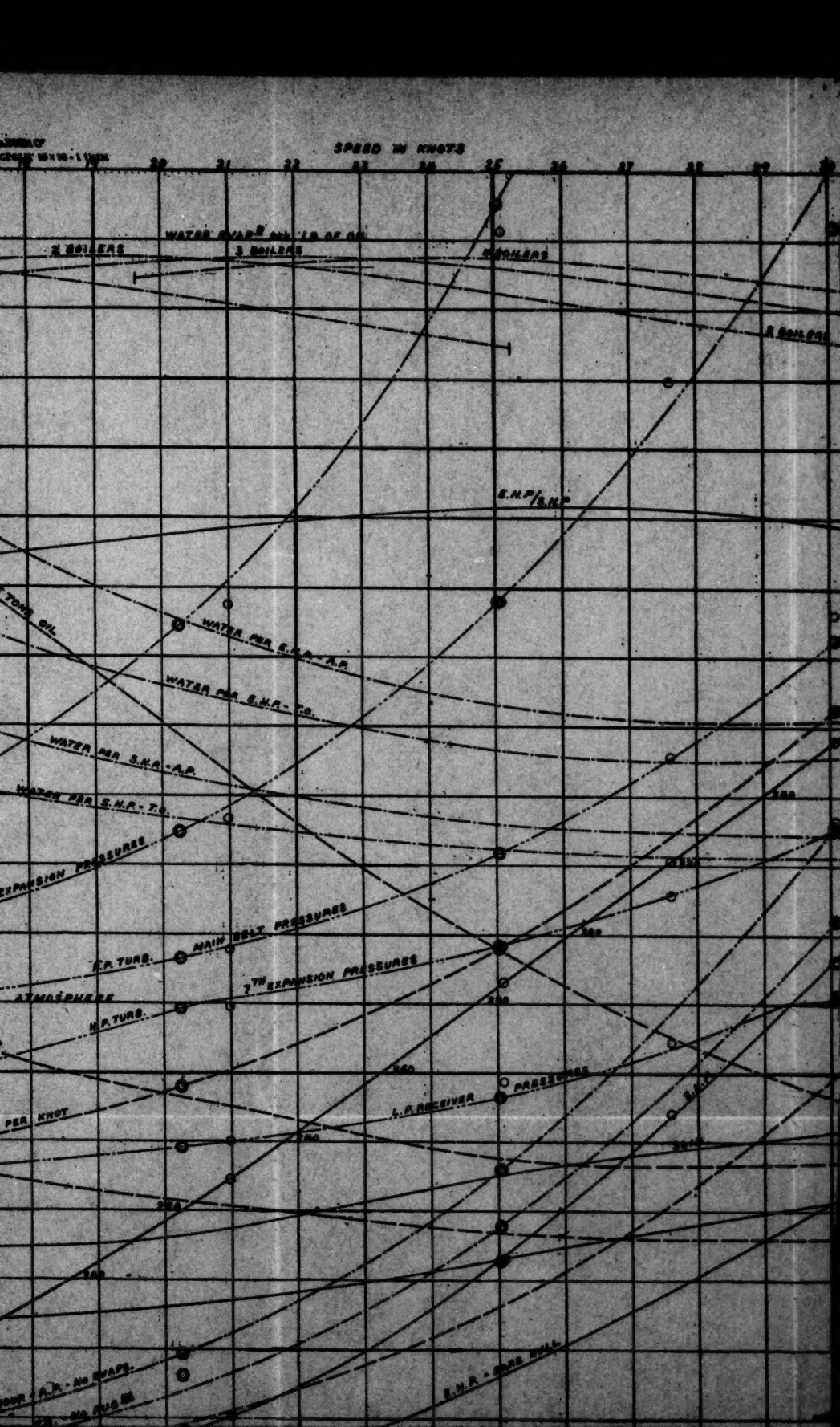
19,500 B.t.u. fuel oil per S.H.P. per hour, pound..... .80
Water for all purposes per S.H.P. per hour, pounds.... 12.2
Water for turbines only per S.H.P per hour, pounds.. 10.6

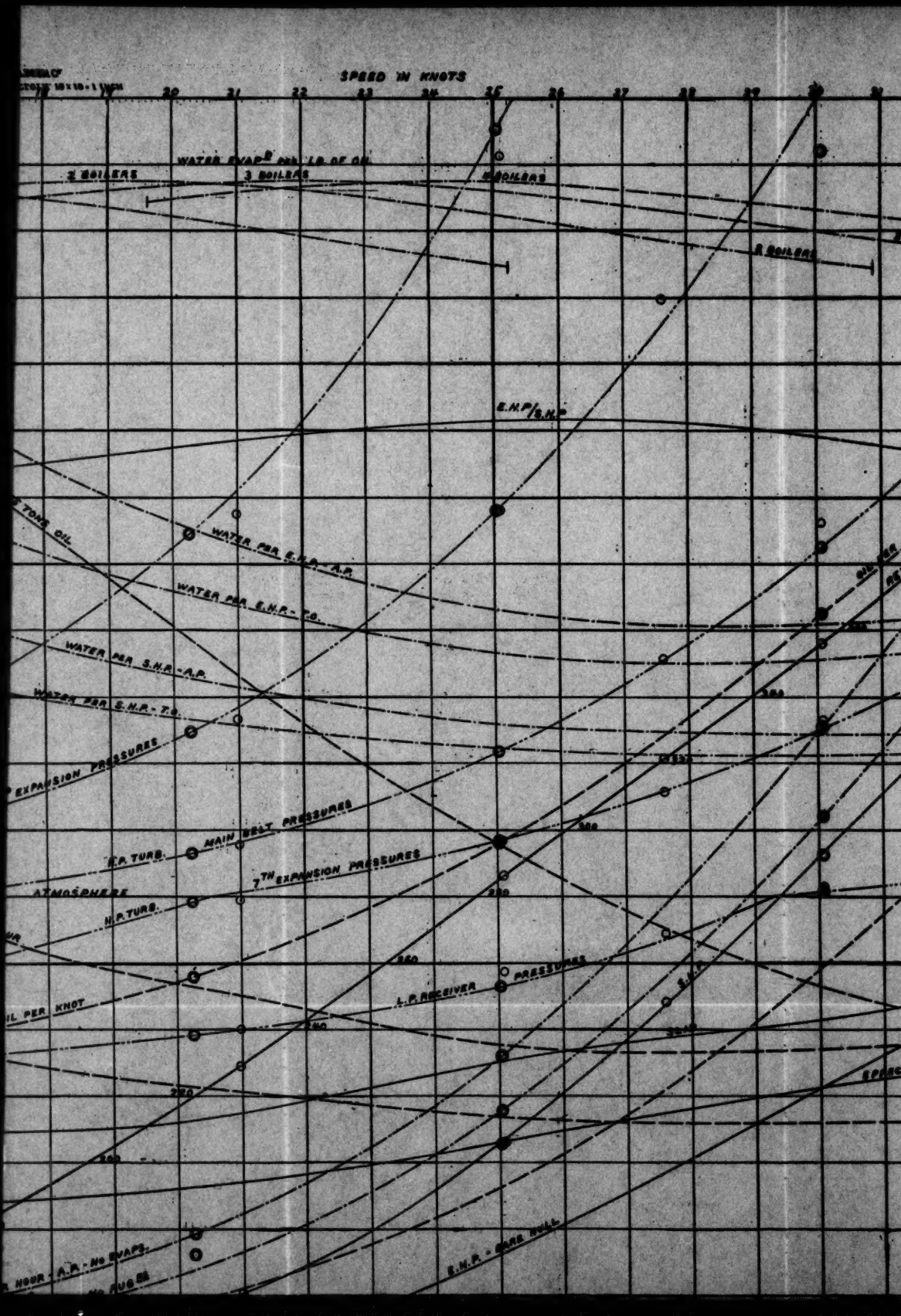
These figures are slightly better than the average obtained on vessels of this class, but they do not represent any marked change since the adoption of the reduction gear in connection with turbine propulsion. Further minor economies can be obtained by the use of superheated steam, feed-water heaters in the uptakes commonly known as "economizers," and more economical turbines. The gain from each is small and the penalties paid in weight, space and loss of reliability are large.

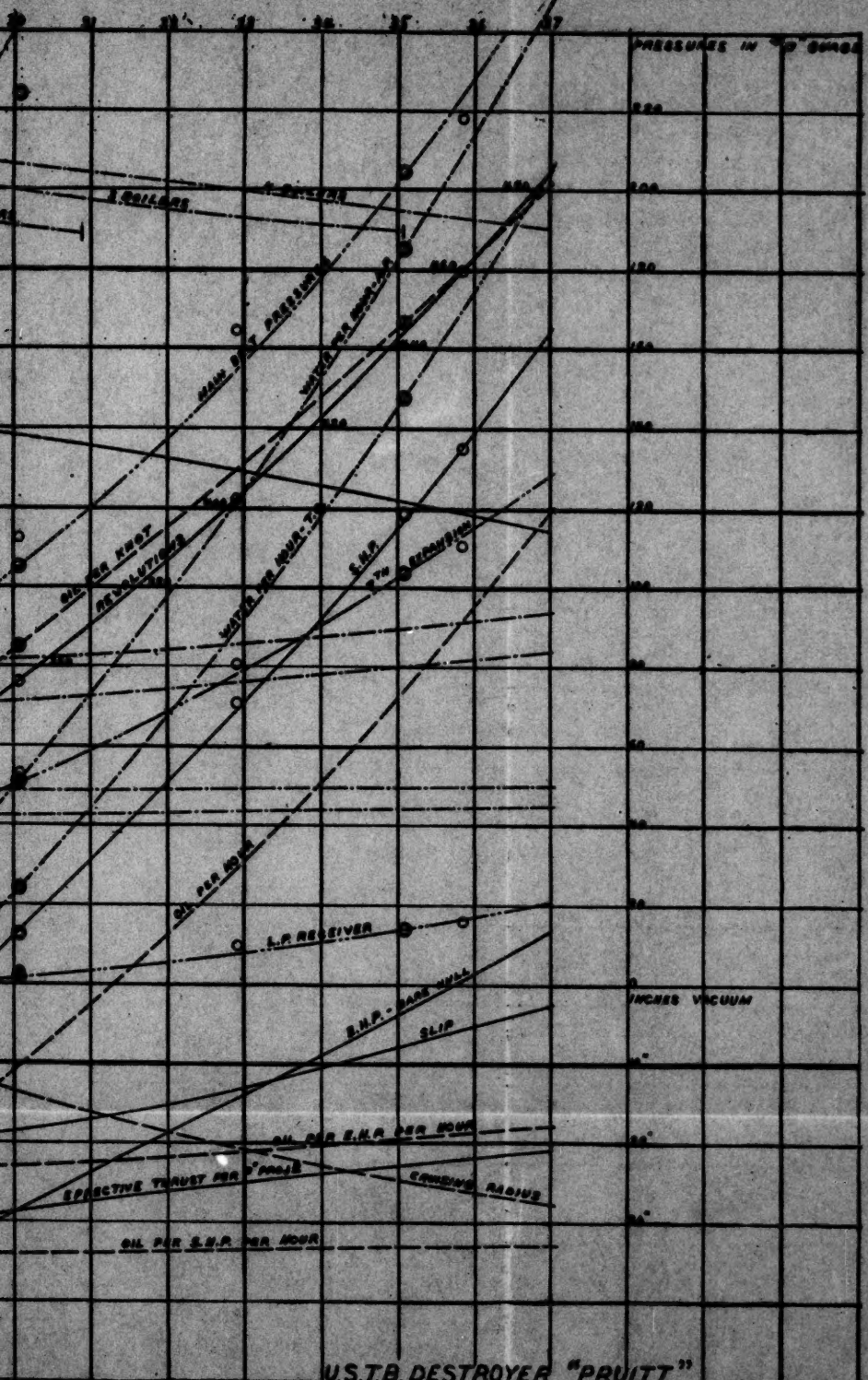
The curve of effective horsepower (bare hull) divided by shaft horsepower, which we usually term, loosely, the curve of propulsive efficiency is typical for a destroyer. The values may be slightly higher than are shown on other types of vessel, but if this is the case, the improvement is due to the reduction of rake of the propelling shafts and great care in the design and construction of appendages such as shaft bosses, struts, circulating water scoops, rudder, and bilge keels.

The curves at the top of the sheet showing the evaporation


WATER PER LB. OF OIL IN LBS (ACTUAL)
WATER PER HOUR IN 1000S LBS
OIL PER HOUR IN 100S LBS
I.H.P. IN 100S
B.H.P. IN 100S
OIL PER KNOT IN LBS
CRUISING RADIUS IN 100S KNOTS
B.H.P./S.H.P IN 1000THS
SLIP IN 10THS PER CENT
EFFECTIVE THRUST IN 10THS #/□" PROJD BLADE
OIL PER H.P. PER HOUR IN 100THS LBS
WATER PER H.P. PER HOUR IN LBS
1 BOILER
CRUISING RADIUS ON 32% TONS OIL
WATER
H.P. TURB. - 1ST EXPANSION PRESS PER
H.P. TURB. - 2ND EXPANSION PR
OIL PER
ATMOSPHERE
B.H.P. PER HOUR
OIL PER B.H.P. PER HOUR
OIL PER KNOT
L.P. RECEIVER
SLIP
EFFECTIVE THRUST PER □" PROJD
REVOLUTIONS
WATER PER HOUR - A.A. - NO


SPEED IN KNOTS
20
21
22
23
24
25
26
27
28
29
30
2 BOILERS
3 BOILERS
4 BOILERS
WATER EVAP. PER LB. OF OIL
E.H.P./S.H.P.
WATER PER S.H.P.-A.P.
WATER PER S.H.P.-T.G.
WATER PER S.H.P.-A.P.
WATER PER S.H.P.-T.G.
EXPANSION PRESSURES
MAIN SELT PRESSURES
L.P. TURB.
7TH EXPANSION PRESSURES
ATMOSPHERE
H.P. TURB.
L.P. RECEIVER PRESSURES
PER KNOT
S.H.P. - SCRS RUN.


SPEED IN KNOTS
20
21
22
23
24
25
26
27
28
29
30
31
2 BOILERS
3 BOILERS
4 BOILERS
WATER EVAP^N PER LB OF OIL
E.H.P./S.H.P.
WATER PER E.H.P. - A.P.
WATER PER E.H.P. - T.E.
WATER PER S.H.P. - A.P.
WATER PER E.H.P. - T.E.
EXPANSION PRESSURES
MAIN BELT PRESSURES
H.P. TURB.
ATMOSPHERE
7TH EXPANSION PRESSURES
L.P. RECEIVER
PRESSURES
E.H.P. - BARE HULL

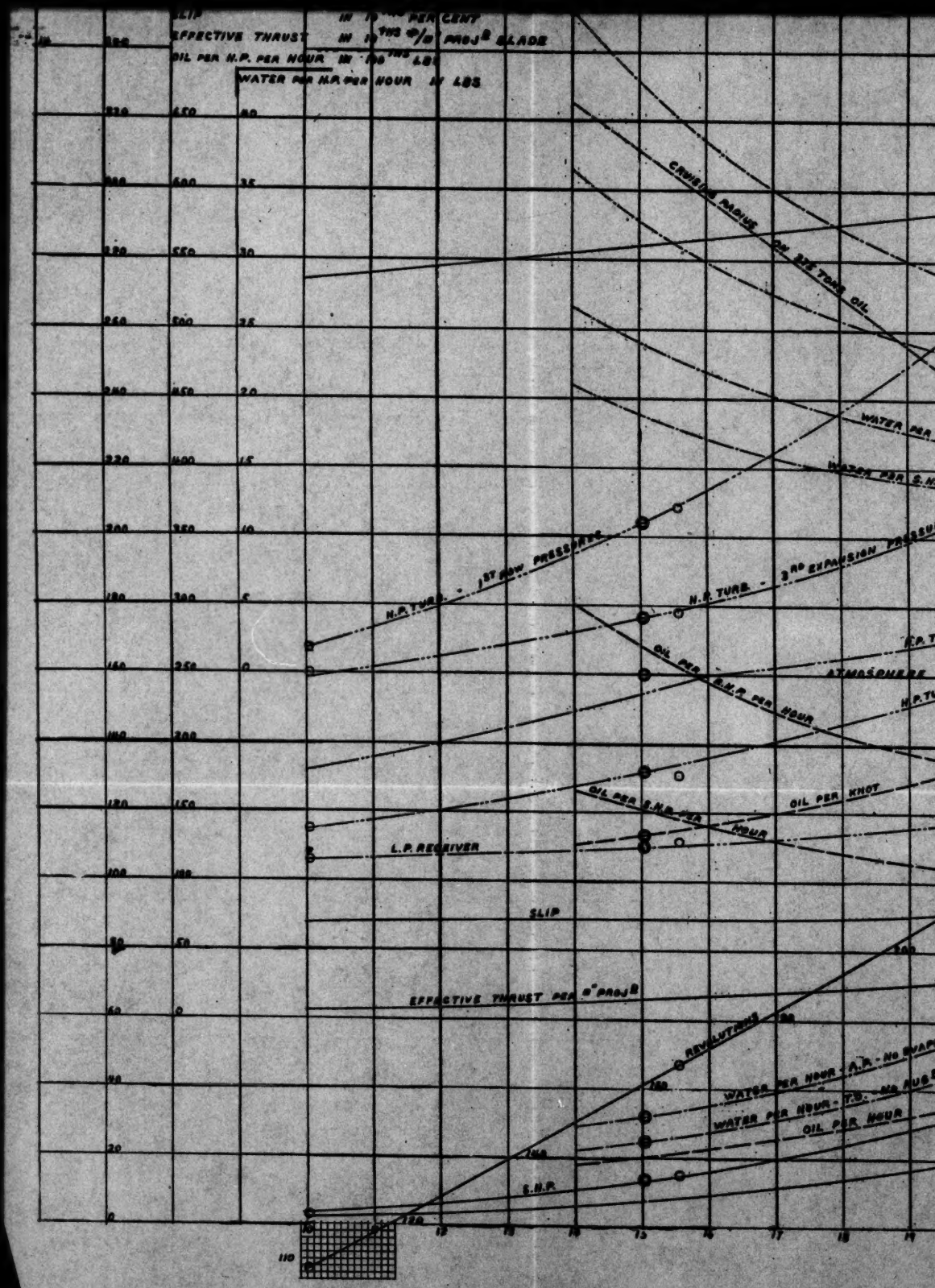


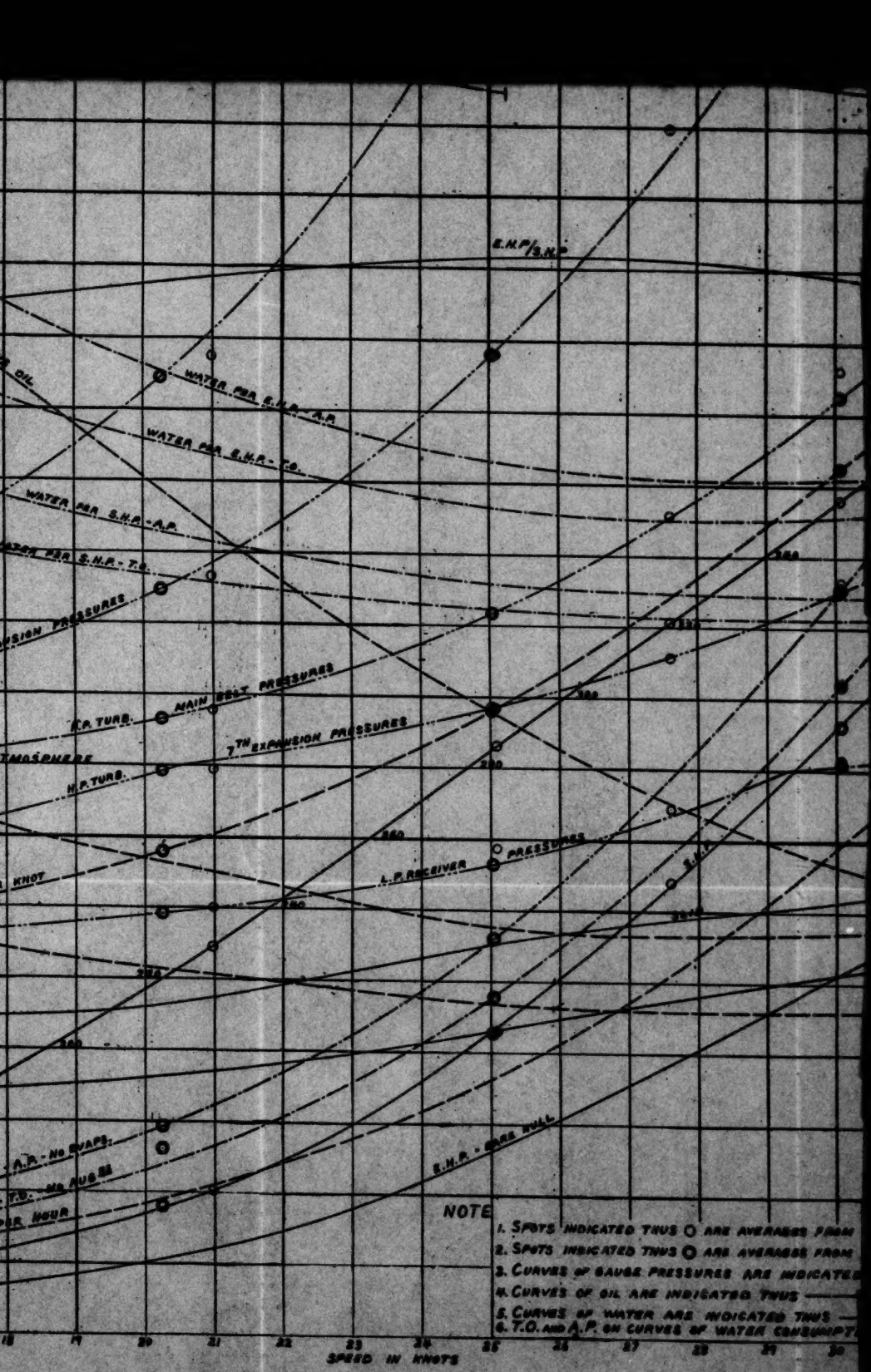
U.S.T.B. DESTROYER "PRUITT"

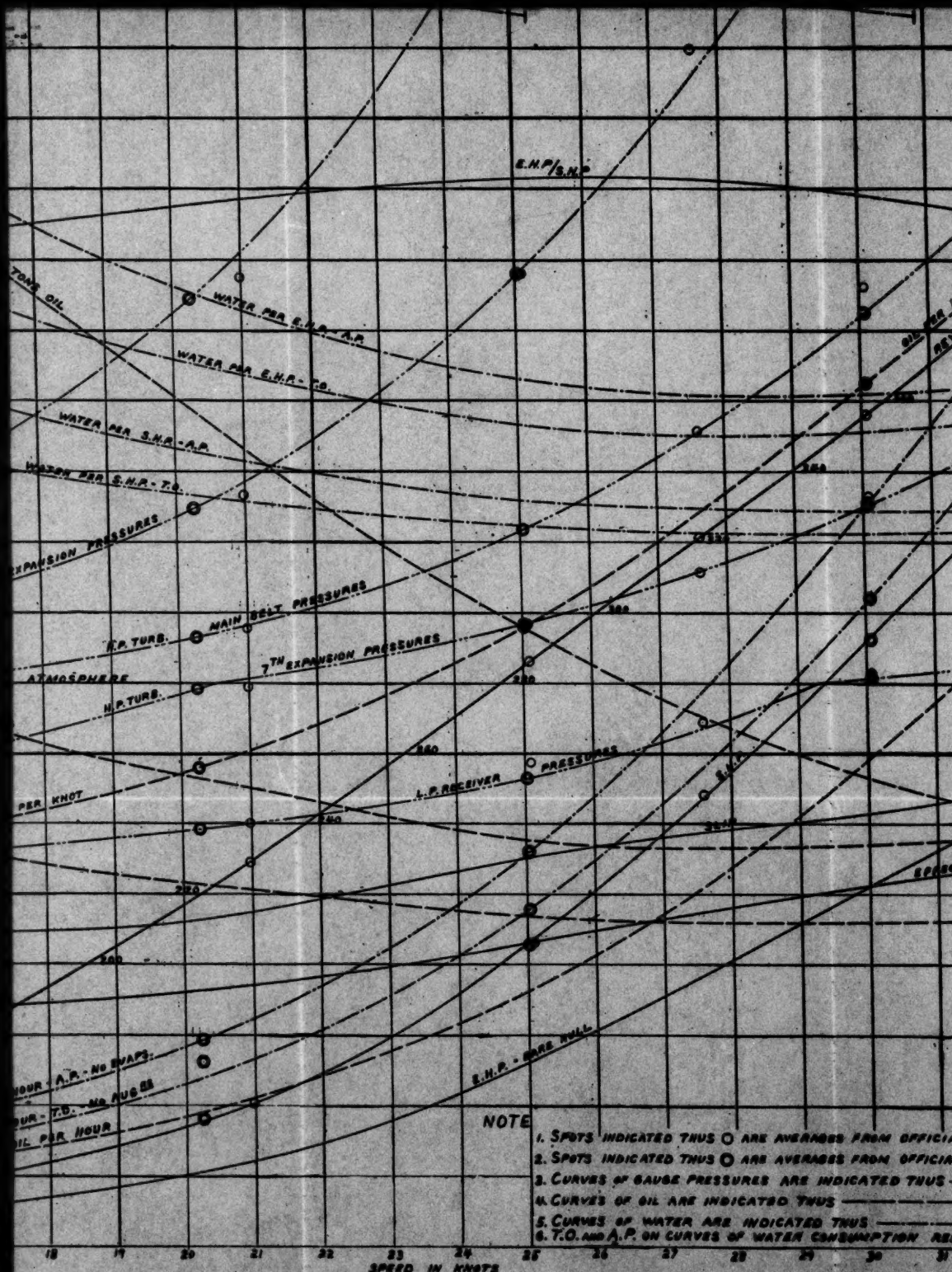
CURVES OF OFFICIAL TRIAL DATA

B.I.W. - DATA

ROCKLAND COURSE - AUG. 19-21, 1920


SLIP IN PER CENT
EFFECTIVE THRUST IN LBS P/D PROJ^R BLADE
OIL PER H.P. PER HOUR IN LBS
WATER PER H.P. PER HOUR IN LBS
CRUISING RADIUS ON 875 TONS OIL
WATER PER
WATER PER S.H.
H.P. TURB. - 1ST FLOW PRESSURE
H.P. TURB. - 3RD EXPANSION PRESSU
H.P. T
OIL PER S.H.P. PER HOUR
ATMOSPHERE
H.P. TU
OIL PER KNOT
OIL PER S.H.P. PER HOUR
L.P. RECEIVER
SLIP
EFFECTIVE THRUST PER 0" PROJ^R
REVOLUTIONS
WATER PER HOUR - A.P. - NO SWAP
WATER PER HOUR - T.D. - NO AUS
OIL PER HOUR
S.H.P.
220
200
180
160
140
120
100
80
60
40
20
0
110
120
13
14
15
16
17
18


E.H.P./S.H.P.
WATER PER E.H.P.-A.P.
WATER PER S.H.P.-T.C.
WATER PER S.H.P.-A.P.
MAIN BELT PRESSURES
7TH EXPANSION PRESSURES
H.P. TURB.
L.P. RECEIVER
PRESSURES
S.H.P.
SPEED IN KNOTS
NOTE
1. SPOTS INDICATED THUS ○ ARE AVERAGES FROM
2. SPOTS INDICATED THUS ○ ARE AVERAGES FROM
3. CURVES OF GAUGE PRESSURES ARE INDICATED
4. CURVES OF OIL ARE INDICATED THUS ——
5. CURVES OF WATER ARE INDICATED THUS —
6. T.C. AND A.P. ON CURVES OF WATER CONSUMPT


E.H.P./S.H.P.
TONS OIL
WATER PER E.H.P. - A.P.
WATER PER E.H.P. - T.O.
WATER PER S.H.P. - A.P.
WATER PER S.H.P. - T.O.
EXPANSION PRESSURES
MAIN BELT PRESSURES
H.P. TURB.
ATMOSPHERE
7TH EXPANSION PRESSURES
H.P. TURB.
PER KNOT
L.P. RECEIVER
PRESSURES
S.H.P.
E.H.P. - BARE HULL
NOTE
1. SPOTS INDICATED THUS O ARE AVERAGES FROM OFFICI
2. SPOTS INDICATED THUS O ARE AVERAGES FROM OFFICIA
3. CURVES OF GAUGE PRESSURES ARE INDICATED THUS -
4. CURVES OF OIL ARE INDICATED THUS ——
5. CURVES OF WATER ARE INDICATED THUS ——
6. T.O. AND A.P. ON CURVES OF WATER CONSUMPTION RE
18
19
20
21
22
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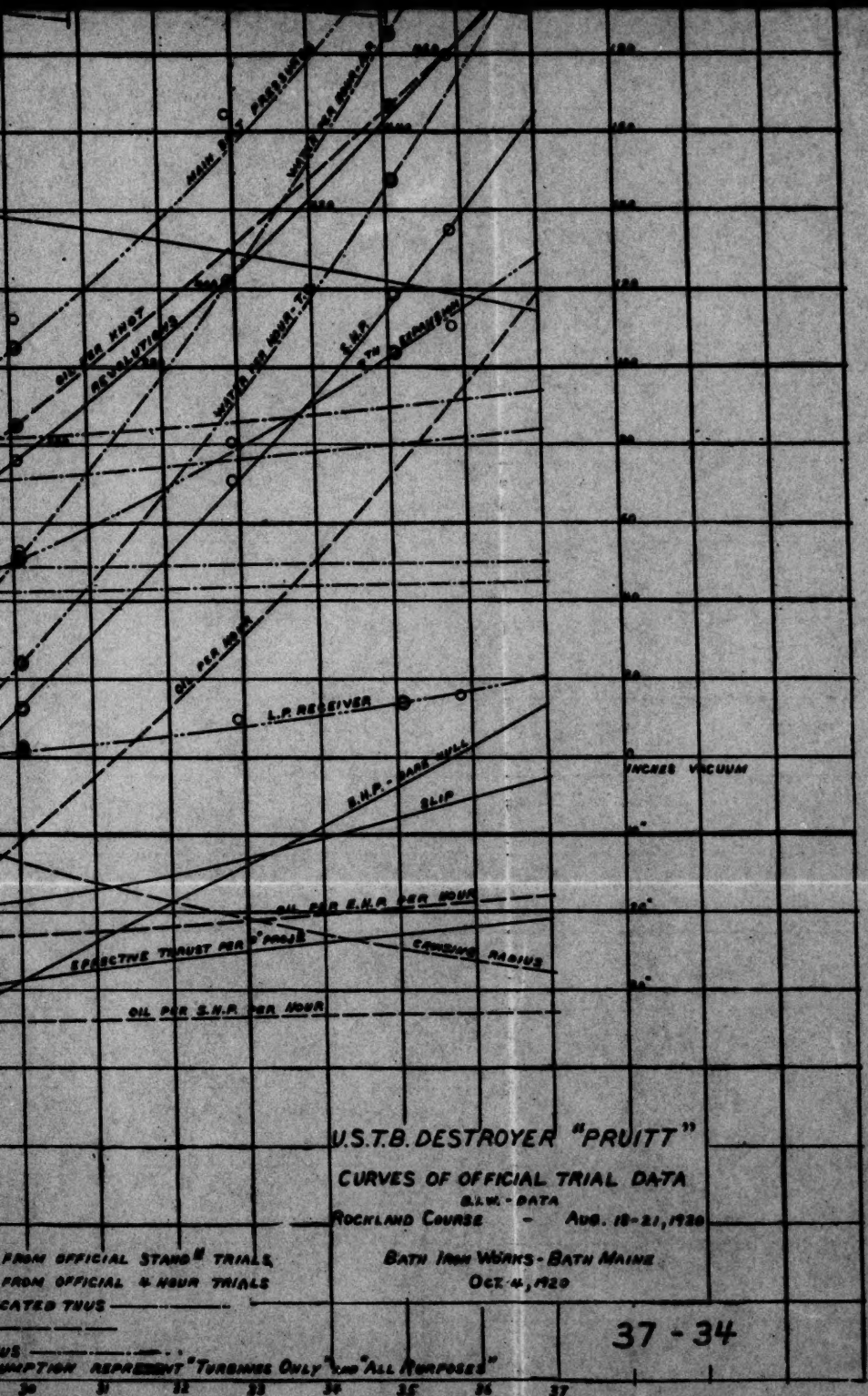
27
28
29
30
SPEED IN KNOTS



U.S.T.B. DESTROYER "PRUITT"

CURVES OF OFFICIAL TRIAL DATA

S.I.W. - DATA

ROCKLAND COURSE - AUG. 18-21, 1920

BATH IRON WORKS - BATH MAINE

OCT. 4, 1920

FROM OFFICIAL STAND[H] TRIALS

FROM OFFICIAL 4 HOUR TRIALS

CATED THUS

US

UMPTION REPRESENT "TURBINES ONLY" AND "ALL PURPOSES"

37 - 34

of water per pound of oil burned are much higher than have been obtained with smaller boilers of this same type on shipboard. The use of large boiler units seems to carry boiler efficiency with it.

It will be noted that—

One boiler will drive this vessel up to 25 knots.

Two boilers will drive this vessel up to 30 knots.

Three boilers will drive this vessel up to 35 knots.

Four boilers will furnish more steam than can be passed through the turbines and auxiliaries.

These curves of evaporation also show the boiler combination that will give the best economy at each speed, but unfortunately the low air pressure necessary for these economies gives a very hot fire room and for comfort of operation it is usually desirable to use fewer boilers and higher air pressure than the combination giving the highest efficiency.

PROPELLER THRUST MEASUREMENTS ON U. S. S. *PRUITT.*

By John E. Burkhardt,

Designing Engineer, Bath Iron Works, Ltd.

The design of marine propelling machinery presents many problems for the solution of which the designer depends upon his ability to obtain and analyze the data taken during the actual performance of similar installations. Measurements of indicated horsepower in the case of reciprocating engines and shaft horsepower for turbine machinery have furnished a basis sufficiently satisfactory for the determination of the principal dimensions of the main engines.

For the determination of the power required to drive the ship we have the results of the model tank experiments, generally expressed as effective horsepower.

The ratio between effective and shaft horsepower, often referred to as the propulsive efficiency, has to be taken from actual trial data and the large variation in the value of this ratio for different types of ships has always presented somewhat of a problem to the designer.

This ratio includes at least four variables:

1. Appendage resistance.
2. Thrust deduction.
3. Wake.
4. Propeller efficiency.

The first three of these have been examined more or less thoroughly in model tank hull experiments but the data is not at all complete. The fourth, propeller efficiency, has been the

subject of extensive systematic model experiments by Admiral D. W. Taylor and Mr. R. E. Froude and much data is available thereon.

The propulsive efficiency, composed of the above factors, is affected by both hull and propeller design, and the desirability of obtaining separate full scale data on each of the above components is manifest.

A knowledge of the actual thrust values would divide the problem into two parts and so provide a partial solution and with this in view an attempt to measure propeller thrust on the *Pruitt* was determined upon.

A description of the vessel and the trial results are the subject of another paper by Mr. Charles P. Wetherbee, in this issue of the JOURNAL, and from this paper it will be seen that the main propelling machinery consists of two sets of turbines and gears on two shafts. On each shaft is the gear wheel driven by one H.P. and one L.P. turbine placed on opposite sides of the gear wheel. Between each pinion and the turbine driving it is a flexible claw coupling, the sleeve of which has a total float of about ½ of an inch. The object of this coupling is to prevent any axial thrust being transmitted from the turbine to the pinion or vice versa. The gearing is of the double helical type, so that the pinion under load will always work in the same fore and aft position relative to the gear wheel. Each turbine has a thrust bearing necessary for its own axial adjustment.

The main shaft is continued through the gear to the main thrust bearing which is of the Kingsbury type located on the forward side of the gear. The housing of the thrust bearing is cast integrally with the gear bed plate. The thrust apparatus itself is applied to the thrust bearing by removing the cover plate on the end and substituting the apparatus.

DESCRIPTION OF THE APPARATUS.

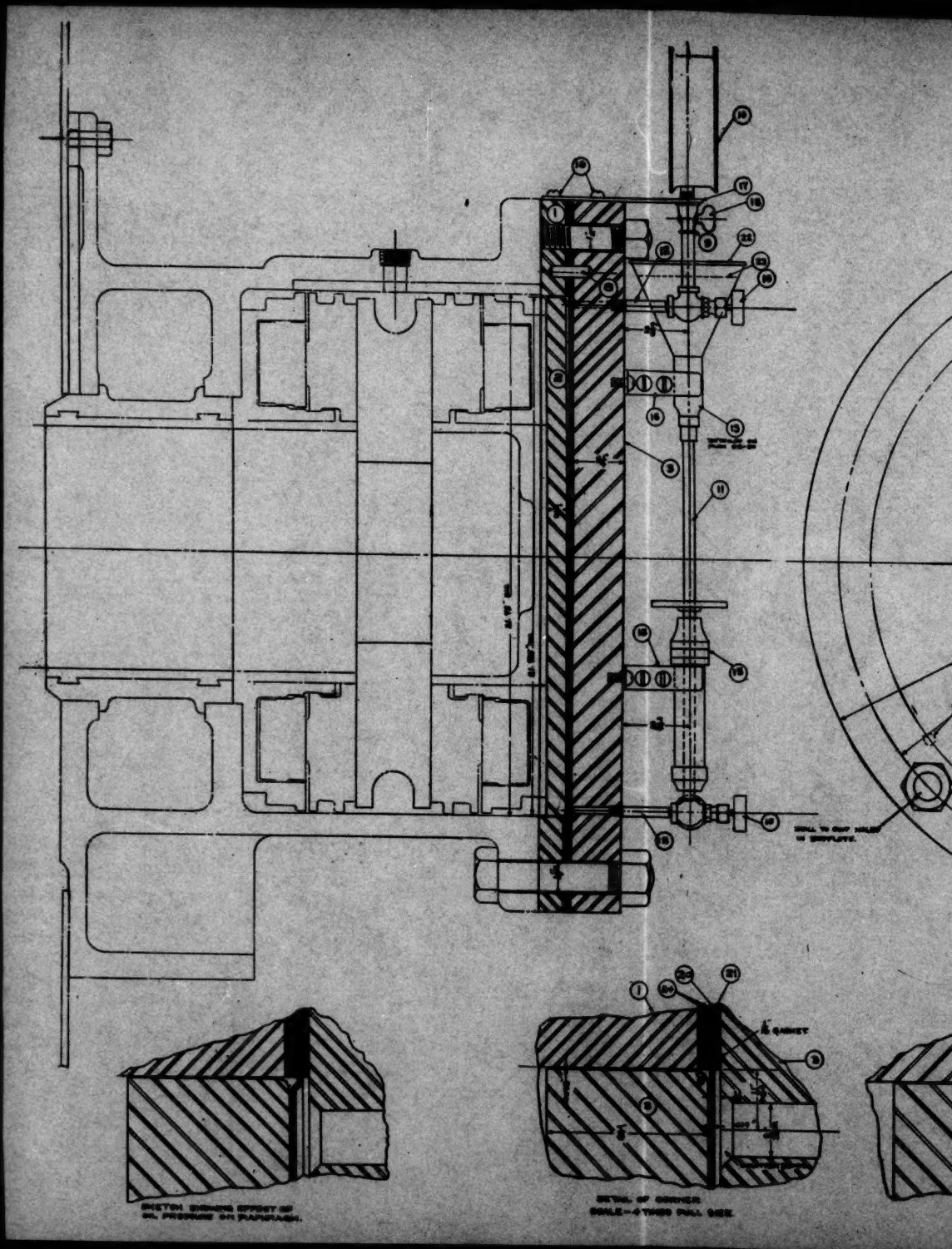
The thrust measuring apparatus is shown on Fig. 1 and consists of a copper diaphragm .019 of an inch thick supported by

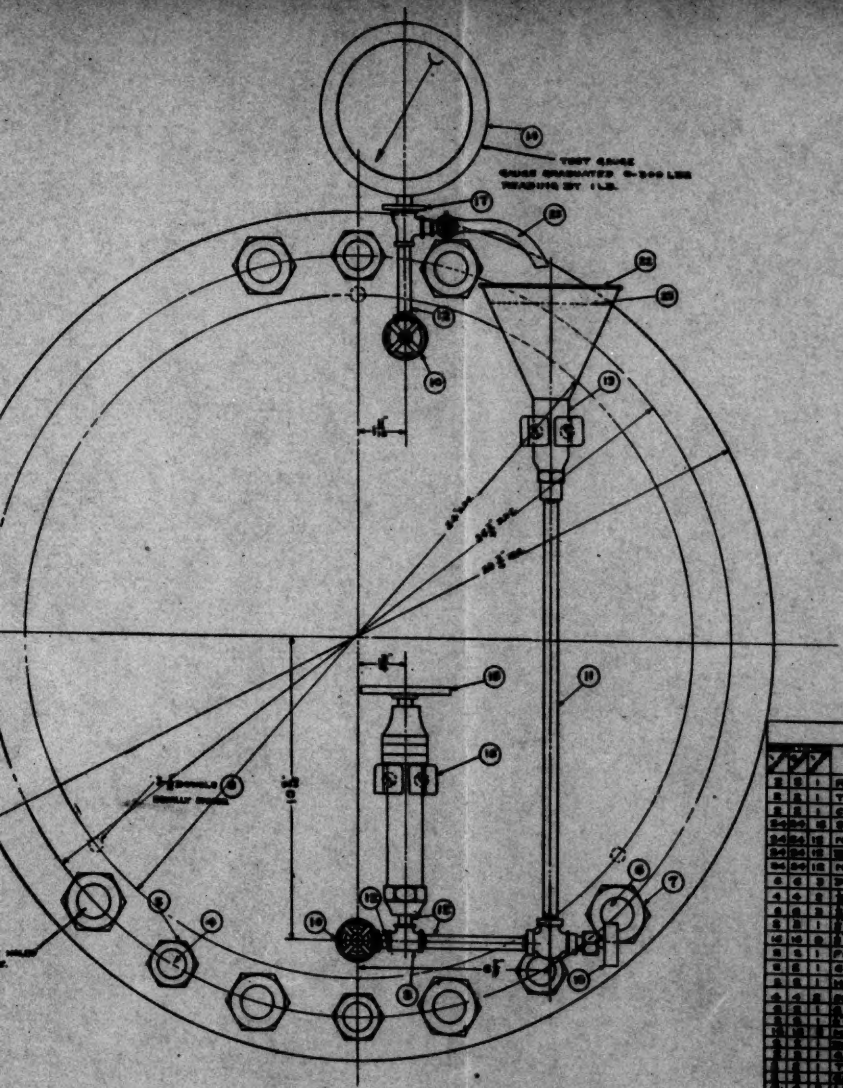
a heavy steel plate with a gasket .041 of an inch thick between the diaphragm and the plate—all three forming a narrow space about 21¾ inches diameter into which oil can be pumped. A clamping ring together with adjusting liners and studs complete the main portion of the apparatus.

As the forward cage containing the levers behind the Kingsbury thrust pads has a large hole to clear the nut on the shaft, it was necessary to interpose a plate marked (2) to distribute the pressure over the diaphragm. The liners marked (24) are for the purpose of making fore and aft adjustments to provide sufficient working oil clearance for proper operation.

The apparatus for handling the oil consists of a hand screw pump, similar to that used on pressure gauge testing machines, funnel, pipe and valves as shown on the drawing. For measuring the oil pressure behind the diaphragm, a sensitive test gauge, graduated to one pound, is fitted as shown on plan. A valve for throttling vibrations of the gauge pointer and a leak off for the release of air and excess pressure into the funnel is also provided. When not in use the oil is released from behind the diaphragm allowing it to close on the heavy cover plate which then takes the thrust as it would if no measuring device were fitted.

To operate for measurements oil is pumped into the space between the diaphragm and cover plate until no further rise in pressure takes place, a condition found to correspond to a displacement of the shaft of rather more than 1/32 inch. Care was necessary to avoid too much pumping which would cause the thrust collar to bear on the astern pads of the thrust bearing, thus giving a false reading of the pressure. The pumping arrangements, however, were such that the work of pumping was somewhat laborious and there was very little actual danger of the operator overdoing it, except perhaps at very low pressures.

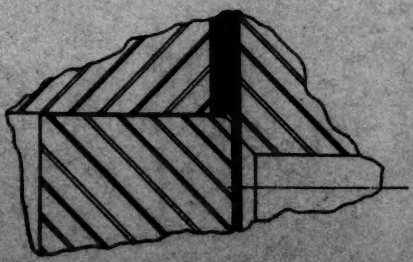

PISTON SHOWING EFFECT OF
OIL PRESSURE ON PACKINGS.
DETAIL OF GUIDER.
SCALE—4 TIMES FULL SIZE.



BILL OF MATERIAL

			NAME	MAT'L
2	2	1	RING	1
3	3	1	THRUST PLATE	2
3	3	1	COVER PLATE	3
24	20	12	STUDS	4
24	24	12	NUTS	5
24	24	12	BOLTS	6
24	24	12	NUTS	7
6	6	3	DOWELS	8
4	4	2	T TEE	9
2	2	2	ANGLE STOP VALVE	10
2	2	1	T PIPE	11
10	10	2	NIPPLES	12
2	2	1	FILLING CUP	13
2	2	1	GAUGE	14
2	2	1	HAND PUMP	15
4	4	2	SUPPORTS	16
2	2	1	GAUGE SUPPORT	17
2	2	1	YOKE	18
2	2	1	BEARING ADJUSTER	19
2	2	1	BEARING	20
2	2	1	ADJUSTING SCREW	21
1	1	1	THRUST	22

FIG. 1



SECTION SHOWING EFFECT OF THRUST ON DIAPHRAGM.

U.S.T.B. DESTROYER 344

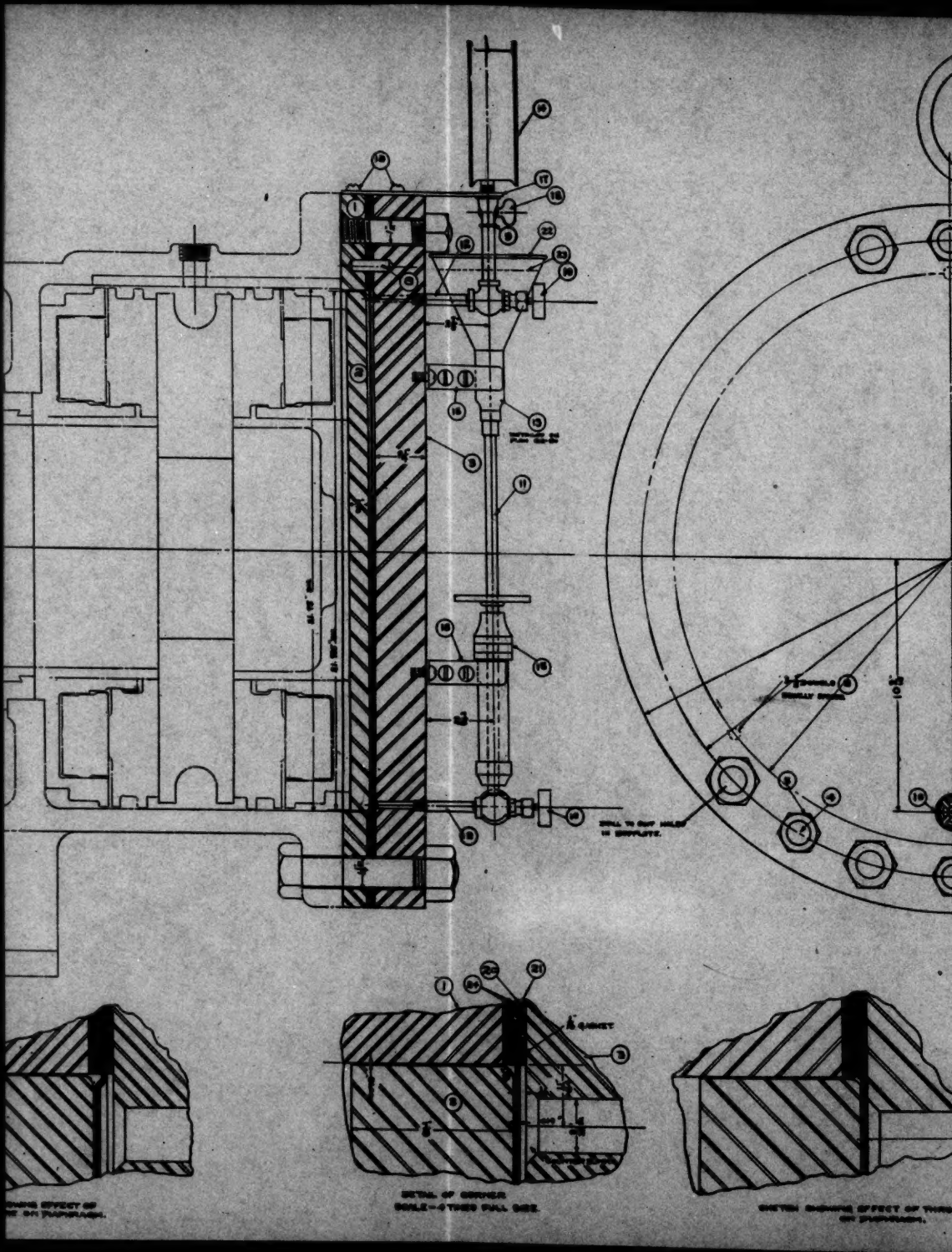
REDUCTION GEAR

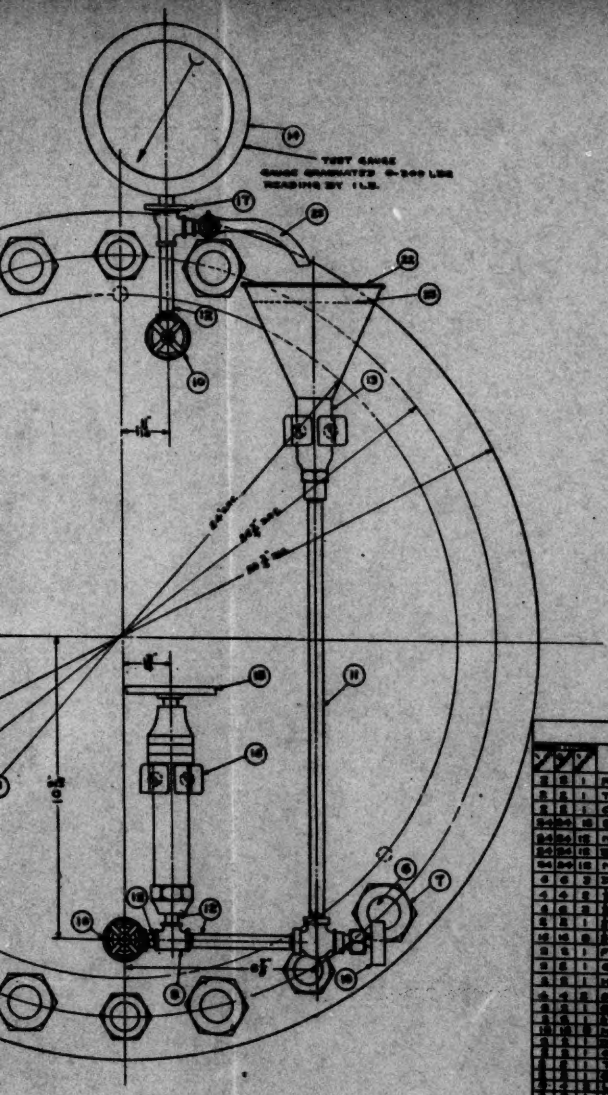
ARRANGEMENT AND DETAILS OF THRUST MEASURING ATTACHMENT

BATH IRON WORKS, BATH, ME.

SCALE - 6" = 1 FOOT.

DATE - JAN 2, 1920





BILL OF MATERIAL

			NAME	ITEM NO	MATERIAL		SOURCE FROM
2	2	1	RING	1	STEEL CL. "C"		STOCK
2	2	1	THRUST PLATE	2	STEEL CL. "B"		STOCK
2	2	1	COVER PLATE	3	STEEL CL. "C"		STOCK
24	24	16	STUDS	4	BOLT MAT. CL. "B"	4218 H	STOCK
24	24	16	NUTS	5	BOLT MAT. CL. "C"	4218 H	STOCK
24	24	16	BOLTS	6	BOLT MAT. CL. "B"	4218 H	STOCK
24	24	16	NUTS	7	BOLT MAT. CL. "B"	4218 H	STOCK
4	4	2	DOWELS	8	BOLT MAT. CL. "B"	4218 H	STOCK
4	4	2	TEE	9	BRASS		STOCK
4	4	2	ANGLE STOP VALVE	10	BRASS		STOCK
2	2	1	PIPE	11	BRASS		STOCK
10	10	5	NIPPLES	12	BRASS		STOCK
2	2	1	FILLING CUP	13	COMPO "M"	4218 H	STOCK
2	2	1	GAUGE	14	BRASS CASE		STOCK
2	2	1	HAND PUMP	15	BRASS		STOCK
4	4	2	SUPPORTS	16	STEEL CL. C		STOCK
2	2	1	GAUGE SUPPORT	17	STEEL CL. "C"		STOCK
2	2	1	SCREW	18	BRASS		STOCK
10	10	5	PACKING GLAND	19	BRASS		STOCK
2	2	1	BONNET	20	BRASS		STOCK
1	1	1	SOCKET - .45"	21	BRASS		STOCK
1	1	1	TUNNEL	22	BRASS		STOCK
1	1	1	SOCKET EXTENSION	23	BRASS		STOCK
2	2	1	CAP - .45"	24	BRASS		STOCK
1	1	1	PIPE	25	BRASS		STOCK

FIG. 1

U.S.T.B. DESTROYER "347"

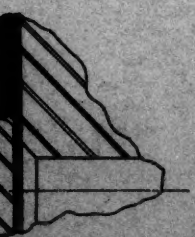
REDUCTION GEAR

ARRANGEMENT AND DETAILS OF THRUST MEASURING ATTACHMENT

SMITH IRON WORKS, BATH, ME.

SCALE - 6" = 1 FOOT.

DATE - JAN 6, 1920



EFFECT OF THRUST
IN DIFFERENCE.

CALIBRATION OF THE APPARATUS.

It was considered necessary to have some means of checking the apparatus to determine whether the product of the pressure read on the gauge and the area behind the diaphragm represented the load on the diaphragm. For this purpose the apparatus was set with its axis vertical on the floor of the shop and heavy loads applied to a plate resting on the diaphragm. The load consisted of a large propeller, shafting and stern tube, and totalled 45,732 pounds. These weights were weighed on an ordinary weighing machine recently certified correct. Micrometer measurements were carefully taken to determine the actual rise of the load, and also to make sure that the axis of the load was kept in a vertical position.

The result of this calibration shows that the effective diameter of the diaphragm was 21.41 inches instead of 21.78 inches, the actual measurement of the diameter of space behind the diaphragm. This reduction, though small, seems to the writer rather more than might be expected from physical considerations of the conditions at the edge of the diaphragm. The whole correction, however, only amounts to 1.6 per cent of the diameter or a little over 3 per cent of the area. This correction was applied to the actual thrust readings taken on the trials by using the effective diameter, as found by the calibration, in place of the actual measured diameter, thus applying the correction proportional to the load.

It is possible, however, that the correction may not be proportional to the load, but somewhere between that and a constant quantity, and for that reason it would have been desirable to calibrate the diaphragm at various loads covering the range anticipated on the trial. For practical reasons, however, it was impossible to get any more dead load on the apparatus for calibration purposes, the maximum possible being only rather more than half the maximum measured on trial. The calibration, therefore, though correct up to 80 knots, may contain an error of about 1½ per cent at full power.

READINGS AND CORRECTIONS.

In order to try out the apparatus, it was installed on the *Sicard*, a sister ship of the *Pruitt*, preceding her by a few months. This boat was only taken out on one day's preliminary trial, and during that trial numerous readings were obtained which were probably quite accurate as far as the measurement of the thrust was concerned. This trial was an ordinary builder's trial and subject to the usual variations, so that revolutions, speed, and other data were not nearly so accurate or reliable as that obtained on the *Pruitt*.

The results obtained on the *Sicard* have been plotted along with those of the *Pruitt*, but for the above reason no great weight was attached to the former in the fairing of the curves.

The corrections applied to the readings consisted of (1) Gauge Calibration Corrections, (2) Calibration Correction of the Apparatus, (*i.e.*, the effective diameter of the diaphragm), (3) Constant Correction for the Weight of the Shafting, etc., and its own rake, (4) Variable Correction for Weight of Shafting, etc., and change of Trim due to Speed (*i.e.*, squat), (5) Variable Correction for Angularity of Measured Thrust to the Horizontal.

On all trials the squat was observed by means of a long pendulum 11 feet in length, consisting of a round machined steel bob suspended in oil by a cord and constrained to roll between two parallel bars laid in a fore and aft direction. This pendulum had been used on previous occasions and found to be very satisfactory.

RESULTS.

The thrust measurements were taken during the official standardization and endurance trials and therefore under very satisfactory conditions.

The method of calculating the thrust from the actual readings for all runs on the *Pruitt* is indicated in Fig. 2 on which also is plotted the spots of all readings of measured thrust for

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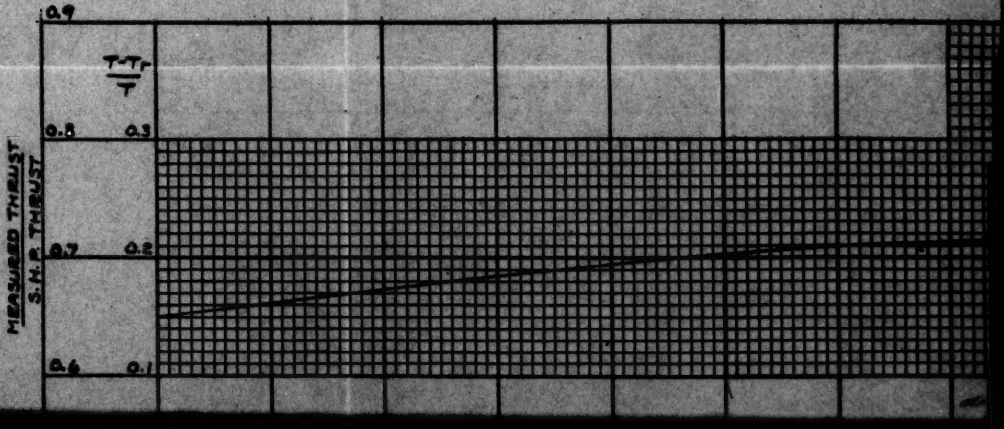
RUN.				REVS. (MEAN)	DISPT.	SPEED	TRIM.				SQUAT		TOTAL SLOPE OF SHAFT (NEET) $\frac{\Theta_T}{L_{PP}+L_B}$	TRUE TRIM $\frac{\Theta_T}{L_{PP}+1}$	SIN Θ_R = TRIM Θ_T - Θ_R IN	AX. COMP. OF WT. OF SHAFT = W SIN Θ_R		REC. THR.
							DRAFT		INCHES	IN/FT	INCHES	IN/FT				PART. W=8159	STRC. W=25800	PART
NO	DATE	BEGIN	END				FORD	AFT										
				RPM	TONS	KNOTS							IN/FT	IN/FT		POUNDS	POUNDS	LBS/D
				R	Δ	V				Θ_R		Θ_R	Θ_T	Θ_R		WAP	WAS	Pp
1-2-3	8-18-20	10.32	11.19	106.7	1235	10.04	9-5	9-7	2	.006	-2½	-.007	.561	.542	.0451	1860	1550	23.0
4-5-6	8-18-20	11.30	12.00	167.1	1234	13.56			2	.006	-3	-.010	.558	.539	.0449	1860	1540	36.5
7-8-9	8-18-20	12.09	12.34	224.2	1233	20.98			2	.006	4½	.014	.582	.562	.0458	1890	1580	66.5
10-11-12	8-18-20	12.43	1.04	284.5	1232	25.09			2½	.007	31½	.101	.670	.647	.0539	2220	1850	119.3
13-14-15	8-18-20	1.12	1.34	321.4	1229	27.69			2½	.007	49½	.160	.729	.704	.0586	2410	2020	158.4
16-17-18	8-18-20	1.43	2.05	364.0	1224	30.08			2½	.008	64½	.209	.779	.752	.0627	2580	2160	186.1
19-20-21	8-18-20	2.14	2.36	402.1	1218	32.91			2½	.008	78	.252	.822	.794	.0661	2720	2270	208.6
22 To 26	8-18-20	2.45	3.26	459.6	1208	35.85	9-2½	9-4½	2½	.008	87½	.282	.852	.823	.0686	2820	2360	230.1
15K	8-20-20	12.43	4.43	161.1	1212	15.02	9-9½	8-11½	-10	-.032	-3	-.010	.520	.502	.0418	1720	1440	19.2
20K	8-19-20	1.48	5.49	220.1	1210	20.25	9-4	9-5½	1½	.004	2½	.009	.575	.555	.0462	1900	1590	62.0
25K	8-21-20	7.20	1.20	285.4	1213	25.08	9-1½	9-7	5½	.017	30½	.098	.678	.655	.0545	2250	1870	98.4
30K	8-20-20	7.23	11.23	356.0	1210	30.08	9-3½	9-5½	2	.006	64	.207	.775	.749	.0624	2570	2120	174.1
E.P	8-19-20	7.33	11.34	443.9	1217	35.08	9-5½	9-6½	½	.002	85	.274	.838	.810	.0675	2780	2320	223.0
DELIVERY	9-2-20			230.0	1153	20.9			0	0	3	.010	.572	.552	.0460	1890	1580	57.5

MEASURED THRUST
S.H.P. THRUST

...WR OF ...F SHIFT ...N Cr	RECORDED THRUST PRESSURES		THRUST CUPS ARE CALIB... OF GAUGES		TOTAL THRUST ON DIAPHRAGM S&A, P		TOTAL THRUST ON PROPELLER ... Vo		ANGULARITY OF SHAFT ...	TAN θ = Q/R	COS θ	FLOW & SHAFT CORRECTIONS (ALL CORRECTED)			CORRECTED THRUST AT 1810 TOWER	
STBD.	PORT.	STBD.	PORT.	STBD.	PORT.	STBD.	PORT.	STBD.								
POUNDS	LBS/□"	LBS/□"	LBS/□"	LBS/□"	LBS.	LBS.	LBS.	LBS.	IN/FT			LBS.	LBS.	LBS.	LBS.	
Wrs	Pp	Ps	Pp	Ps	tp	ts	T'p	Ts	θ			Tp	Ts	Tp'Ts	T	
1550	23.00	25.00	28.5	23.5	7280	8460	9240	10810	.821	.0518	.9987	9230	10000	19230	19000	
1560	36.5	36.5	34.0	35.0	12200	12600	14060	14140	.618	.0515	.9987	14030	14120	28150	27630	
1580	64.5	67.33	64.0	64.8	23060	23350	24940	24930	.440	.0534	.9986	24910	24900	49810	49000	
1850	119.3	113.7	116.8	113.2	42100	40300	44220	42690	.721	.0601	.9982	44040	42570	86610	85300	
2020	158.0	157.5	115.5	159.0	56000	56900	58440	58920	.775	.0646	.9979	58290	58800	117090	115000	
2160	180.8	180.0	178.3	180.0	64200	64300	66780	66960	.822	.0686	.9977	66630	66810	133440	132300	
2270	208.0	246.0	205.5	206.0	74000	74200	76780	76970	.864	.0720	.9973	76510	76760	153270	152000	
2360	230.0	229.5	227.5	229.8	82000	82300	84880	85460	.892	.0743	.9972	84680	84980	169600	169600	
1440	19.25	29.75	24.75	23.25	8920	8380	10630	9880	.584	.0486	.9988	10640	9840	20450	20450	
1590	67.00	67.00	64.5	64.5	23200	23200	25100	24790	.634	.0528	.9986	25060	24760	49820	49280	
1870	98.6	105.3	104.6	105.3	37700	37950	39980	39820	.787	.0668	.9982	39880	39750	79630	79630	
2120	174.2	168.0	180.7	170.0	65100	61200	67670	63320	.880	.0683	.9977	67540	63160	130670	130670	
2320	223.5	220.0	223.0	220	80400	79200	83180	81580	.880	.0733	.9973	82960	81300	164260	163000	
1580	57.5	65.0	63.5	63.5	22850	22850	24830	24430	.631	.0525	.9986	24800	24400	49200	51200	



	CONVEX-THRUST (CORRECTED).	CORRECT THRUST AT 120 R.P.M.	
S.	LBS.	LBS.	
	TONS	T	
0	19230	19000	
0	28150	27620	
00	44840	44000	
70	86810	85300	
00	117090	115000	
10	133940	132300	
60	152770	152000	
20	169500	169500	
0	20450	20450	
20	44820	44820	
20	79630	79630	
20	130670	130670	
20	164260	163 000	
00	49300	51200	

WEIGHT OF SHAFTING IN LBS.

ITEM	PORT	STAR.
PROPELLER	5277	5277
SHAFTING	24189	17419
GEAR WHEEL	8496	8496
2 PINIONS	2890	2890
THRUST BEARING	300	300
TOTAL	41152	34383
	W_p	W_s

RAKE OF SHAFT

VERTICAL = .562°/FT. = α_v

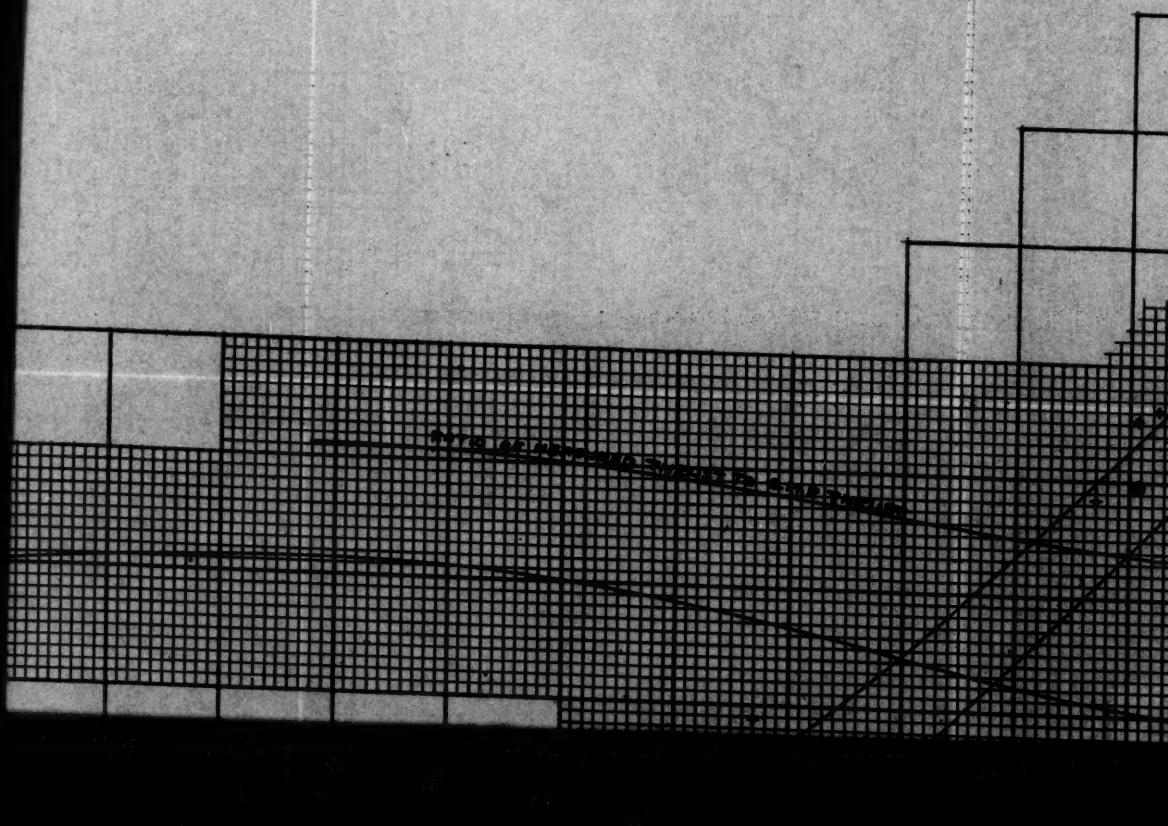
HORIZONTAL = .2665°/FT. = α_h

COMBINED = .622°/FT. = α_c

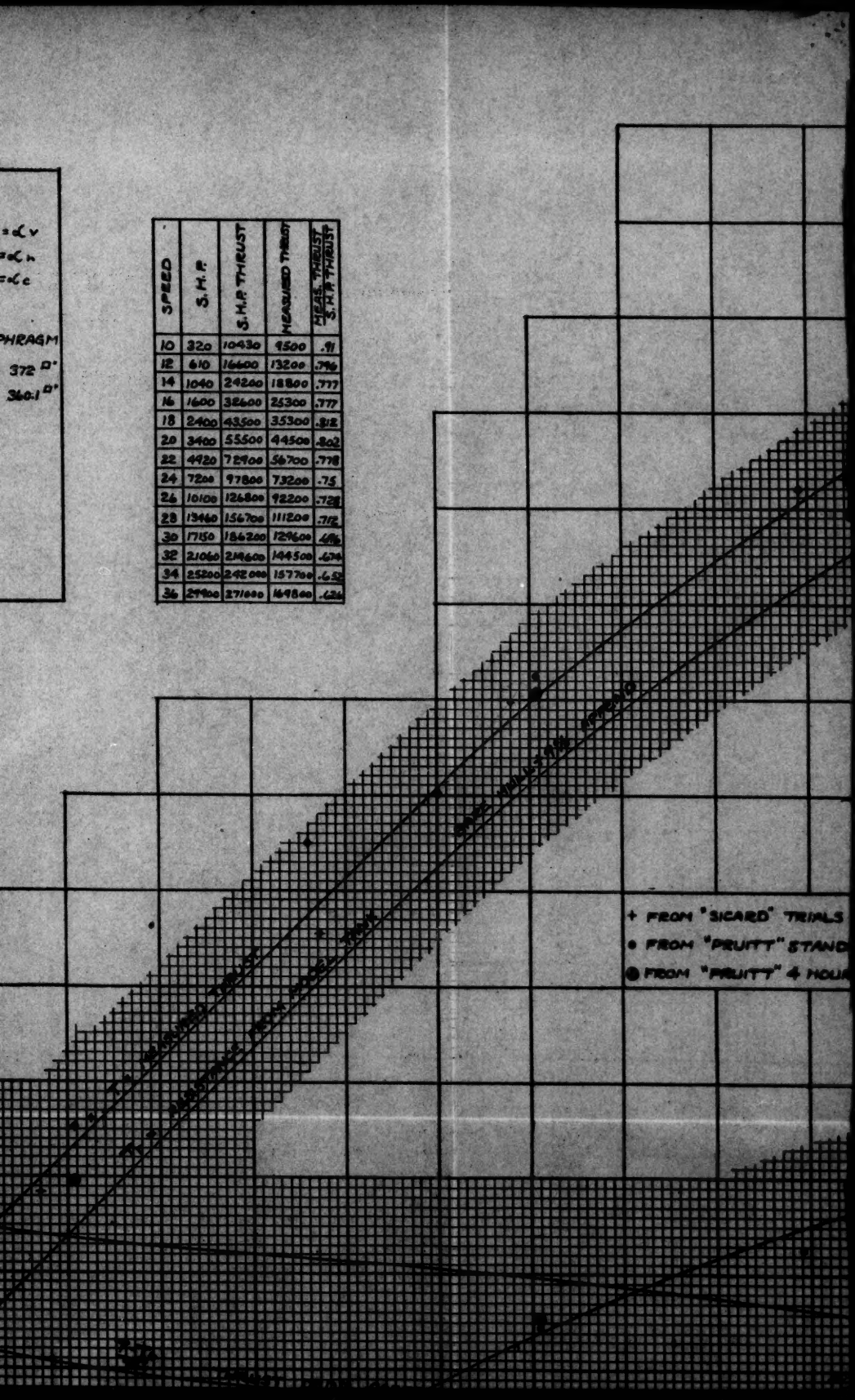
CALIBRATION OF DIAPHRAGM

ACTUAL DIA. 21.78" AREA 372 □"

EFF. DIA. 21.41" AREA 360.1 □"



SPEED	S.H.P.	S.H.P. THRUST	MEASURED THRUST	MEAS. THRUST / S.H.P. THRUST
10	320	10430	9500	.91
12	610	16600	13200	.796
14	1040	24200	18800	.777
16	1600	32600	25300	.777
18	2400	43500	35300	.812
20	3400	55500	44500	.802
22	4920	72900	56700	.778
24	7200	97800	73200	.75
26	10100	126800	92200	.728
28	13460	156700	111200	.712
30	17150	186200	129600	.696
32	21060	214600	144500	.674
34	25200	242000	157700	.652
36	29900	271000	169800	.626



TING IN LBS.

PORT	STAR.
5277	5277
84189	17419
8496	8496
2890	2890
300	300
41152	34383
W_p	W_s

RAKE OF SHAFT

VERTICAL = .562"/FT. = α_v

HORIZONTAL = .2665"/FT. = α_h

COMBINED = .622"/FT. = α_c

CALIBRATION OF DIAPHRAGM

ACTUAL DIA. 21.78" AREA 372 □"

EFF. DIA. 21.41" AREA 360.1 □"

SPEED	S.H.P.	S.H.P. THRUST	MEASURED THRUST	MEAS. THRUST / S.H.P. THRUST
10	320	10430	9500	.91
12	610	16600	13200	.796
14	1040	24200	18800	.777
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18	2400	43500	35300	.812
20	3400	55500	44500	.802
22	4920	72900	56700	.778
24	7200	97800	73200	.75
26	10100	126800	92200	.728
28	13460	156700	111200	.712
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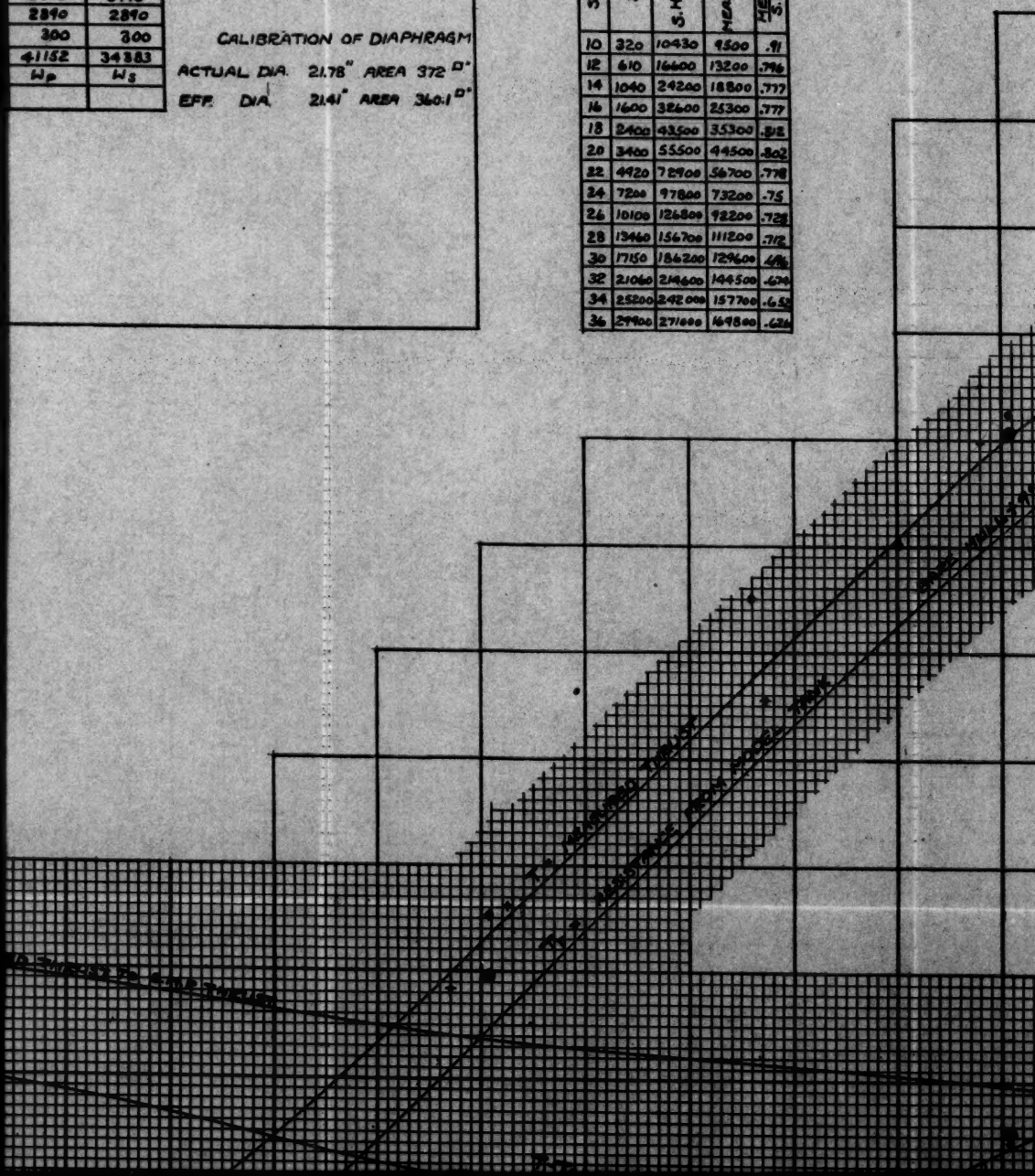
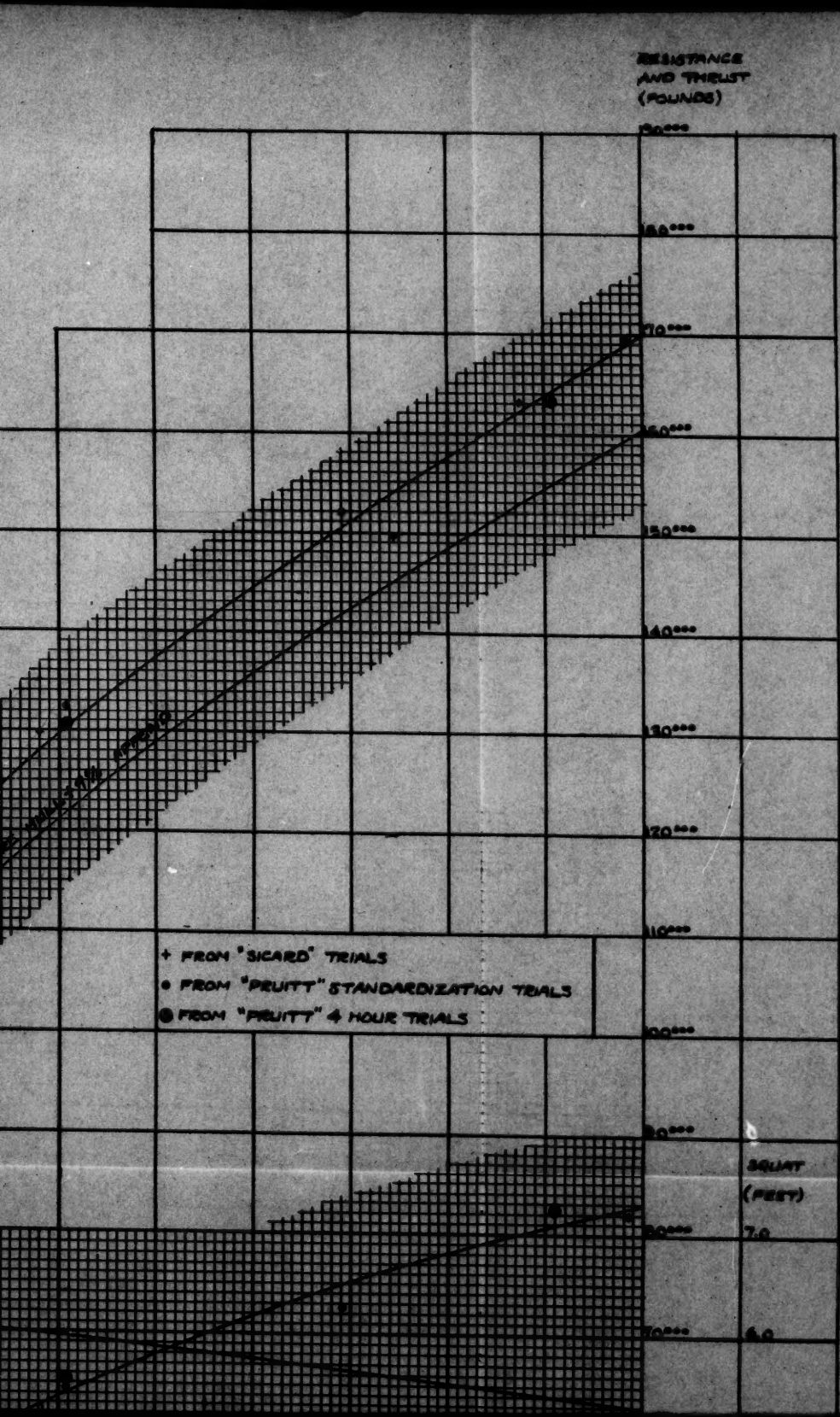
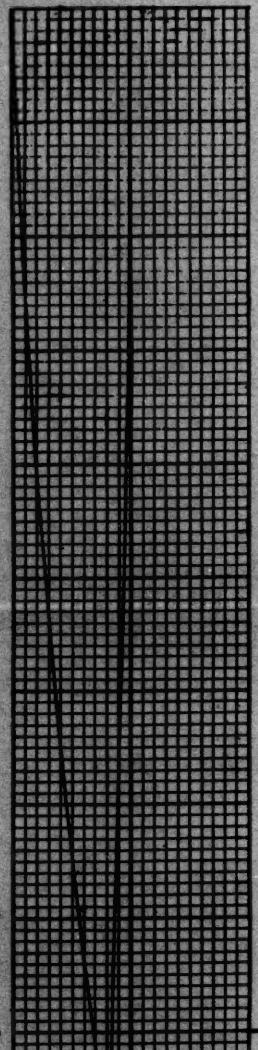
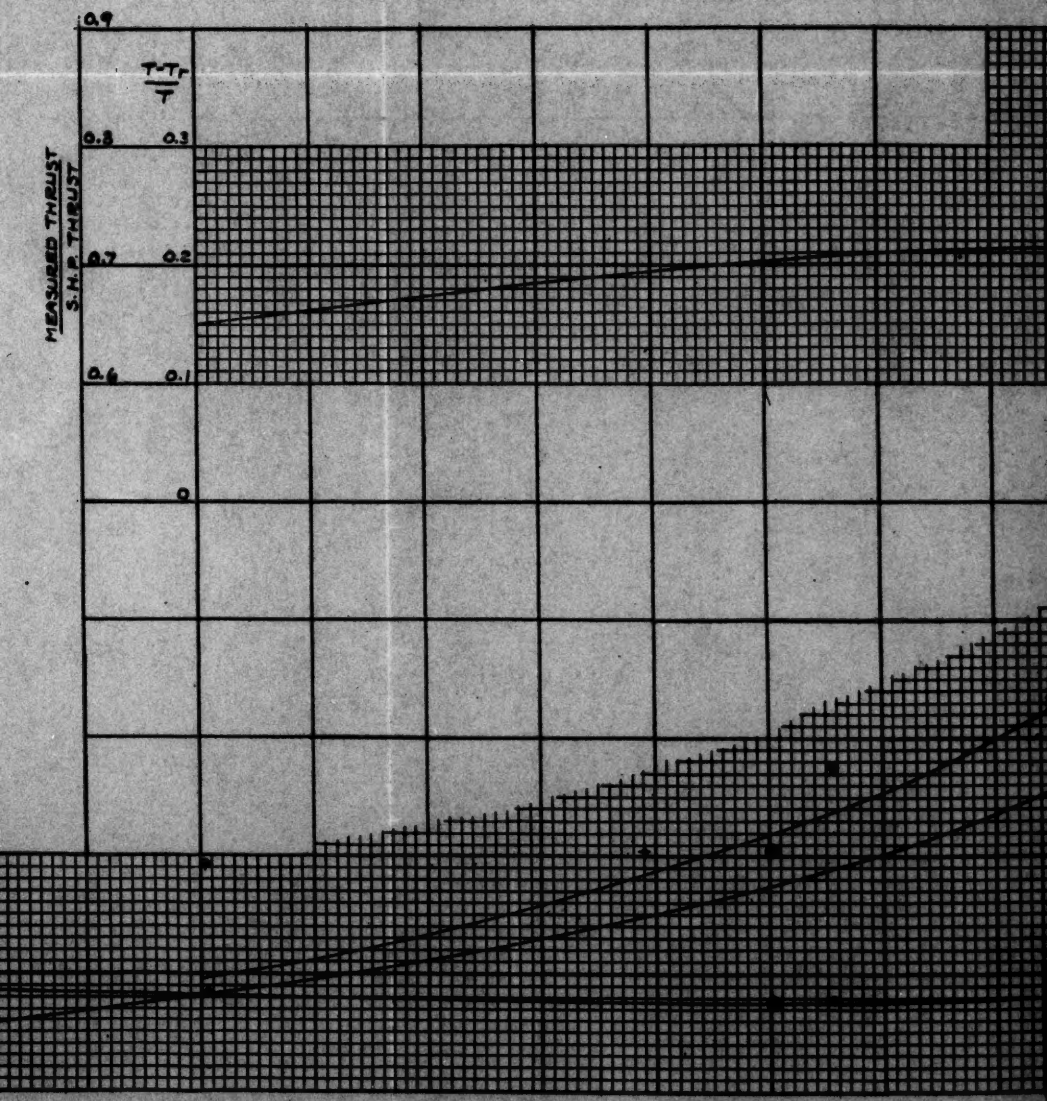
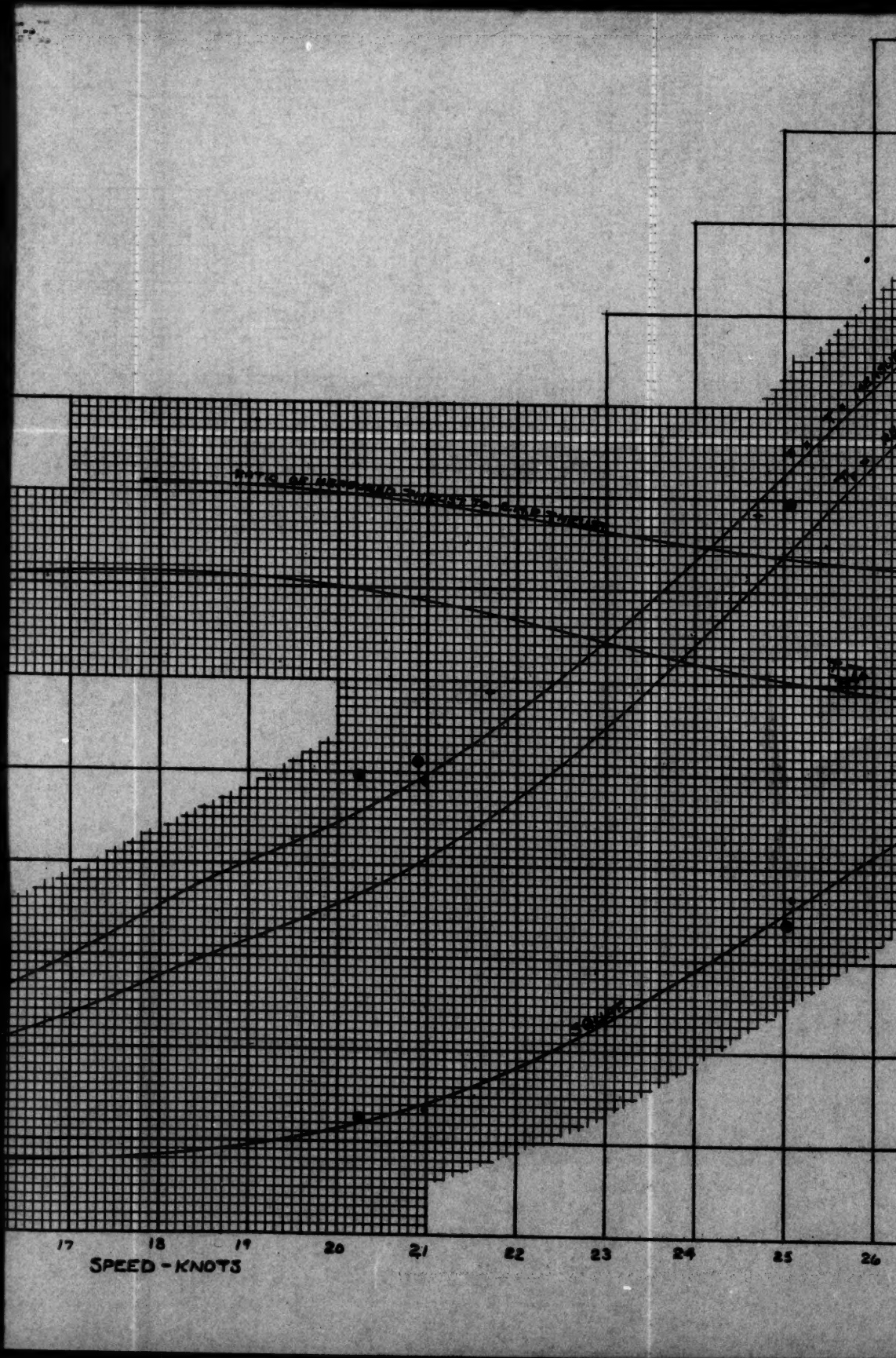



RESISTANCE AND THRUST (POUNDS)
190,000
180,000
170,000
160,000
150,000
140,000
130,000
120,000
110,000
100,000
90,000
80,000
70,000
+ FROM "SICARD" TRIALS
• FROM "FRUITT" STANDARDIZATION TRIALS
● FROM "FRUITT" 4 HOUR TRIALS
SQUAT (FEET)
7.0
6.0



MEASURED THRUST
S.H.P. THRUST


0.9
$\frac{T-T_r}{T}$
0.3
0.8
0.2
0.7
0.1
0.6
0
MEASURED THRUST / S.H.P. THRUST
9
10
11
12
13
14
15
16
17


17
18
19
20
21
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25
26
SPEED - KNOTS

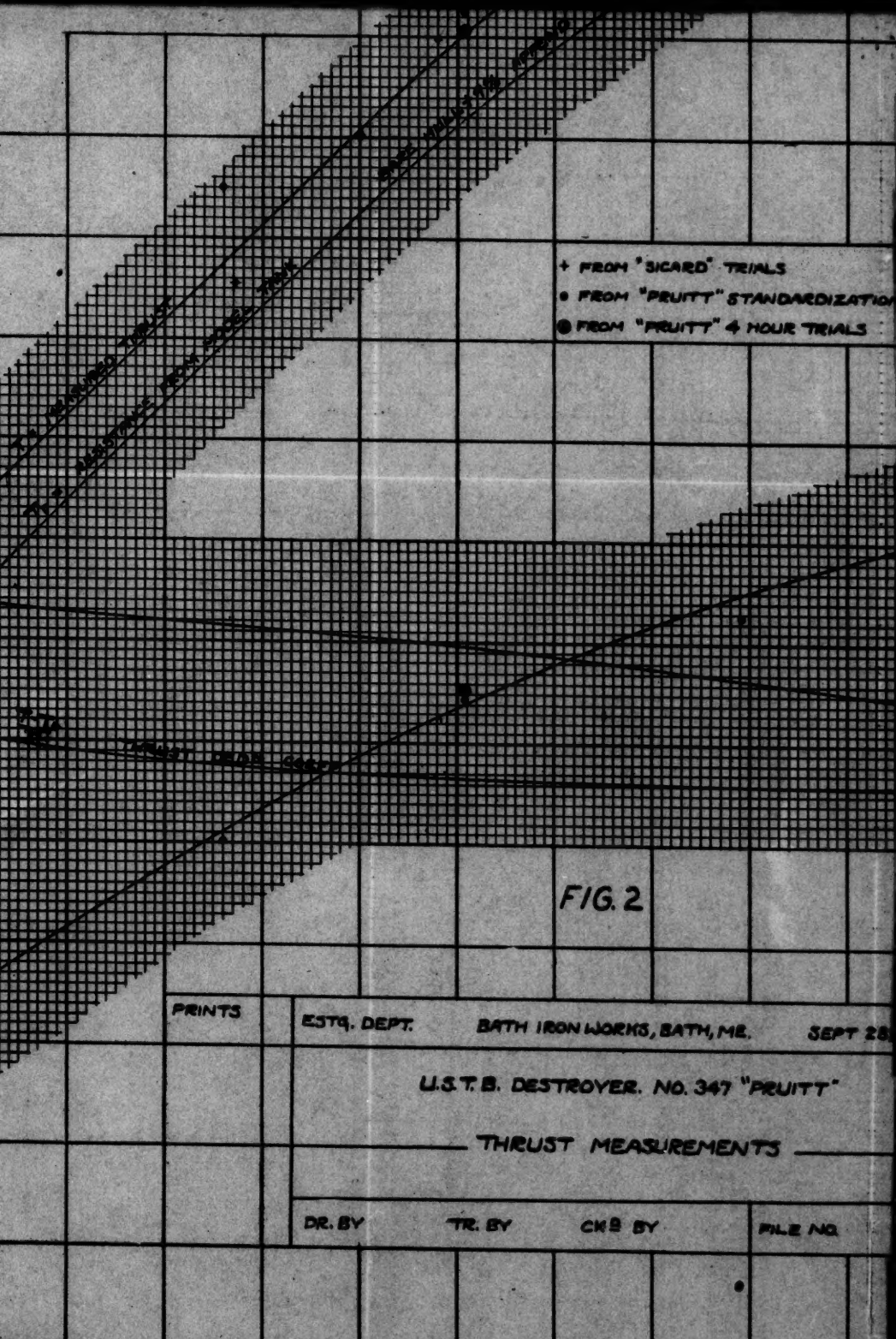

FIG. 2

PRINTS

ESTG. DEPT. BATH IRON WORKS, BATH, ME. SEPT 28

U.S.T.B. DESTROYER. NO. 347 "PRUITT"

THRUST MEASUREMENTS

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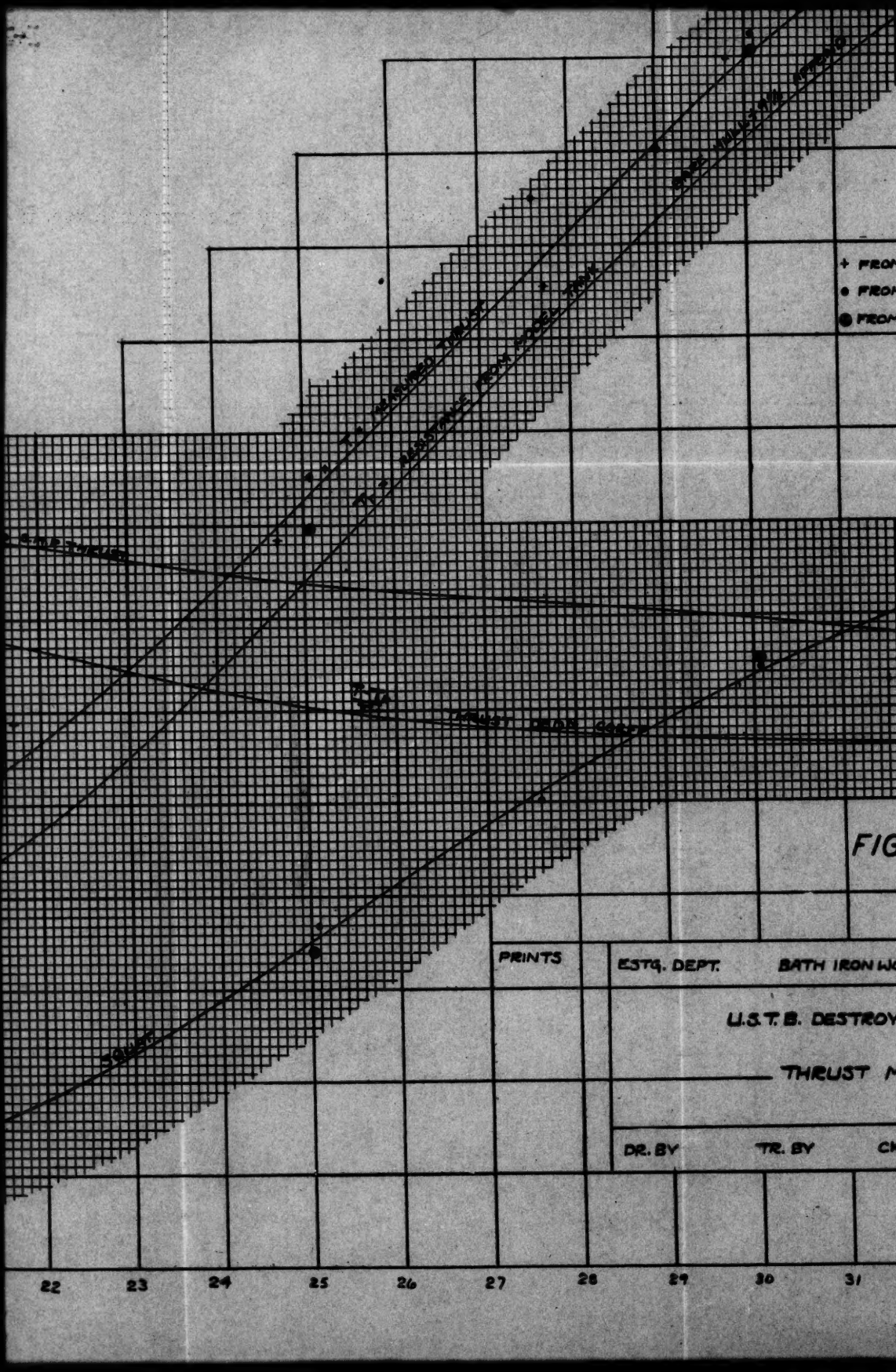
+ FRON
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PRINTS
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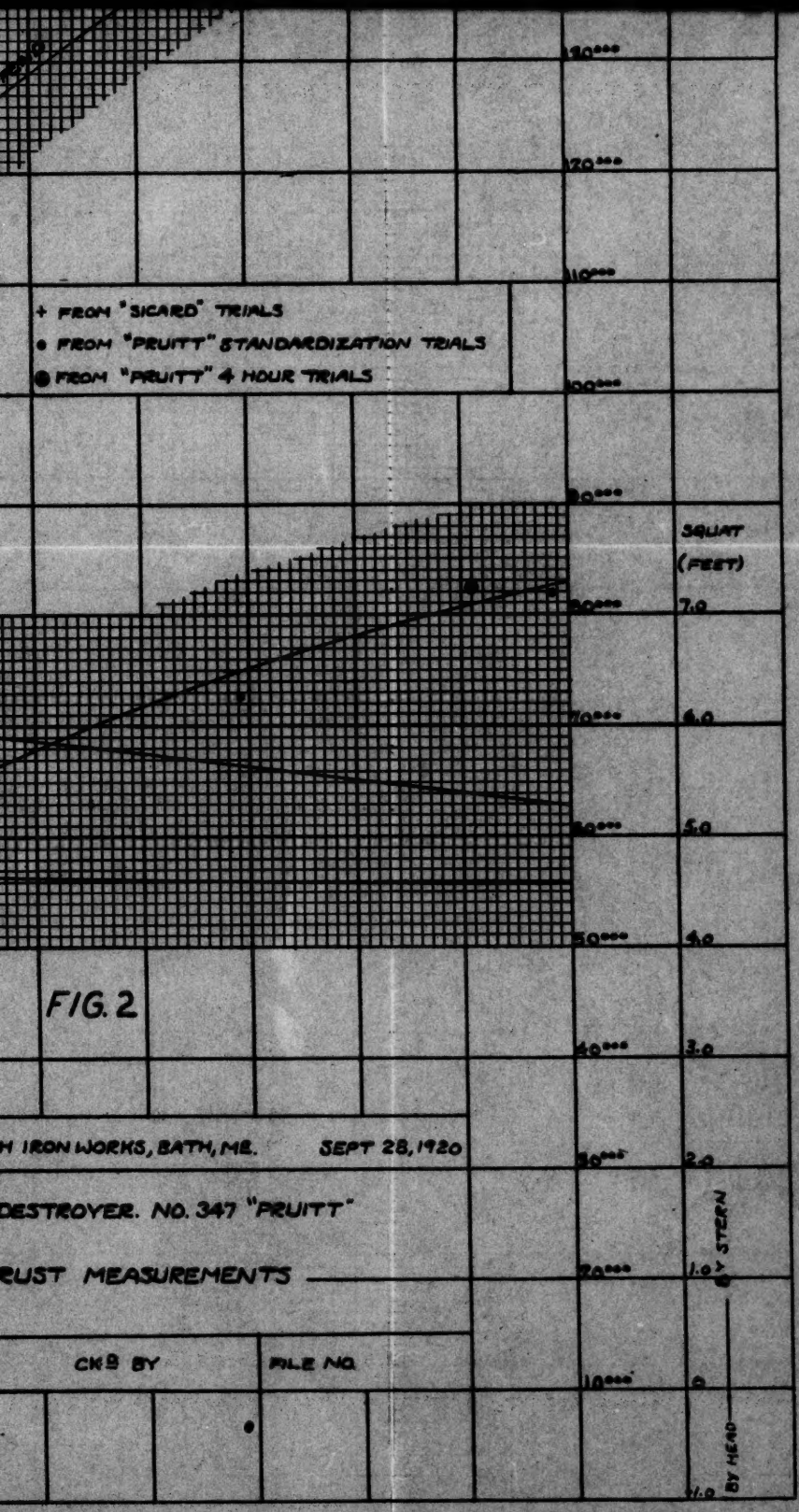


FIG. 2

H IRON WORKS, BATH, ME. SEPT 28, 1920

DESTROYER. NO. 347 "PRUITT"

RUST MEASUREMENTS

CK'D BY

FILE NO.

both *Sicard* and *Pruitt*. The curve of measured thrust here indicated has been drawn through the spots after giving due weight to those considered most reliable. Beside the curve of measured thrusts is plotted the tow rope resistance obtained from the model tank bare hull plus 9 per cent for appendages. The thrust deduction coefficient, taken as equal to $\frac{T-Tr}{T}$ where T is the measured thrust in pounds and Tr is the tow rope resistance including appendages in pounds, has been plotted to the same base, and this thrust deduction coefficient curve was used to some extent in fairing the curve of measured thrust. The effect of this will be noted in the hump in the measured thrust curve in the neighborhood of 18 and 19 knots.

It will be noticed that the thrust deduction coefficient varies from a maximum of .215 at about 18 knots to a minimum of rather less than .06 at maximum speed. It is probable that the falling off in the thrust deduction coefficient at speeds below 18 knots may not be true, because the thrust meter readings below 20 knots are not considered reliable, and there is some question also as to the accuracy of the results at 20 knots.

COMPARISON OF "PRUITT" AND "MACKAY."

Unknown to us until quite recently, contemporary measurements of thrust were made on the British Destroyer Leader *Mackay* and the results were embodied in a paper read before the Institution of Naval Architects by C. Frodsham Holt. The *Mackay* and *Pruitt* were vessels not very different in size and speed and a comparison of results is of interest. Particulars of the vessels are given in the following table:

	Pruitt.	*Mackay.*
Length B.P., feet	310	320
Beam, feet and inches	30–11½	31–9
Draft (load), feet and inches	9–4½	10–6
Displacement	1210	1800
Block coeff. (approx.)	.47	.59 load

The lines differed particularly at the stern, the *Pruitt* having a V stern and the *Mackay* a flat one.

Mr. Holt in his paper deduces values of wake by comparing the propeller performance with the model results of Froude and finding the wake values necessary to make the full size and model results agree. This comparison is made on two independent premises, one based on the S.H.P. and the other on thrust. There is quite a difference in the results obtained by the two methods and moreover the values, ranging from 8 to 25 per cent, are much higher than any previous experience would lead us to expect for such a model as the *Mackay*.

Accordingly the following comparison of *Pruitt* and *Mackay* has been made neglecting wake, that is, in the formula for thrust horsepower, T.H.P. = .00307 TV, V has been taken, for the purposes of this comparison, to be the speed of the ship relative to still water instead of the speed of the ship relative to wake.

As the wake could not be very different in the two ships, this simplification should not appreciably affect the comparison, but due notice should be taken of the substitution when considering the values independently. E.H.P. in each case is from the model tank with an allowance for appendages. In the case of the *Pruitt* this allowance is 9 per cent. S.H.P. was taken from the torsionmeters in both ships. The comparison is shown by curves plotted on a base of speed.

Fig. 3 shows the usual propulsive efficiency, E.H.P./S.H.P.

In Fig. 4 is plotted T.H.P./S.H.P. which may be taken to be a measure of the efficiency of the propeller itself; in fact, if the wake is the same in both vessels this ratio gives an exact comparison, and as will be shown later there is reason to believe that the wakes were similar.

In Fig. 5 E.H.P./T.H.P. has been plotted to indicate a comparison of hull efficiency since the denominator of this ratio involves wake and thrust deduction.

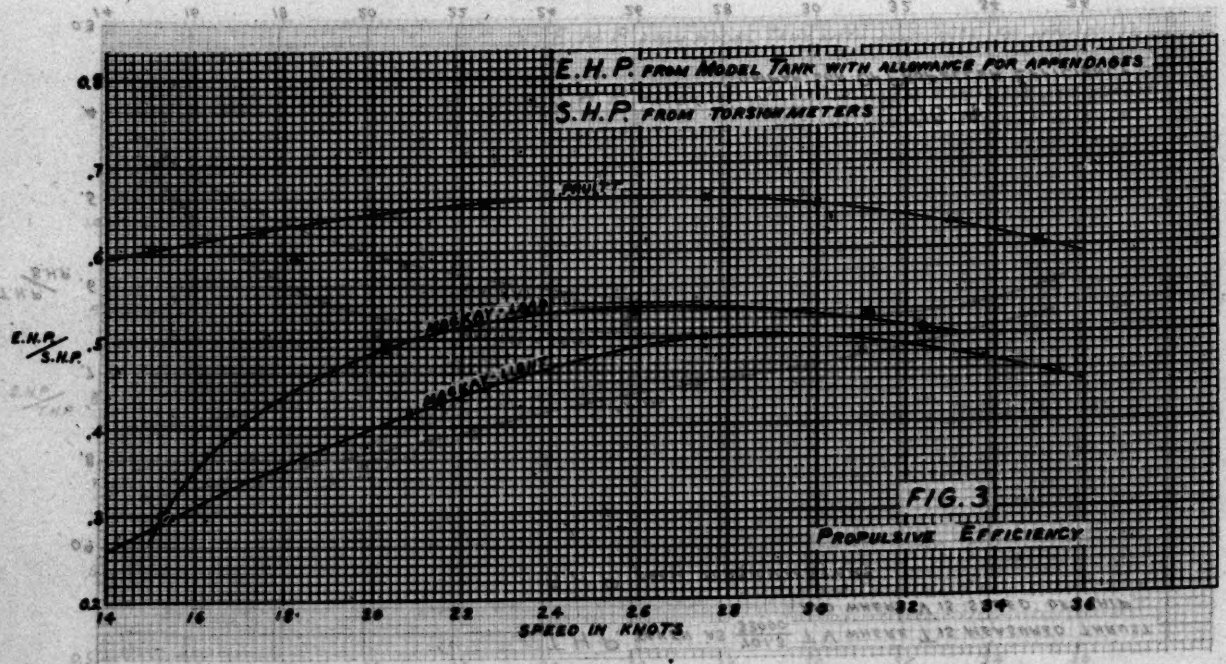


FIG. 3
PROPULSIVE EFFICIENCY

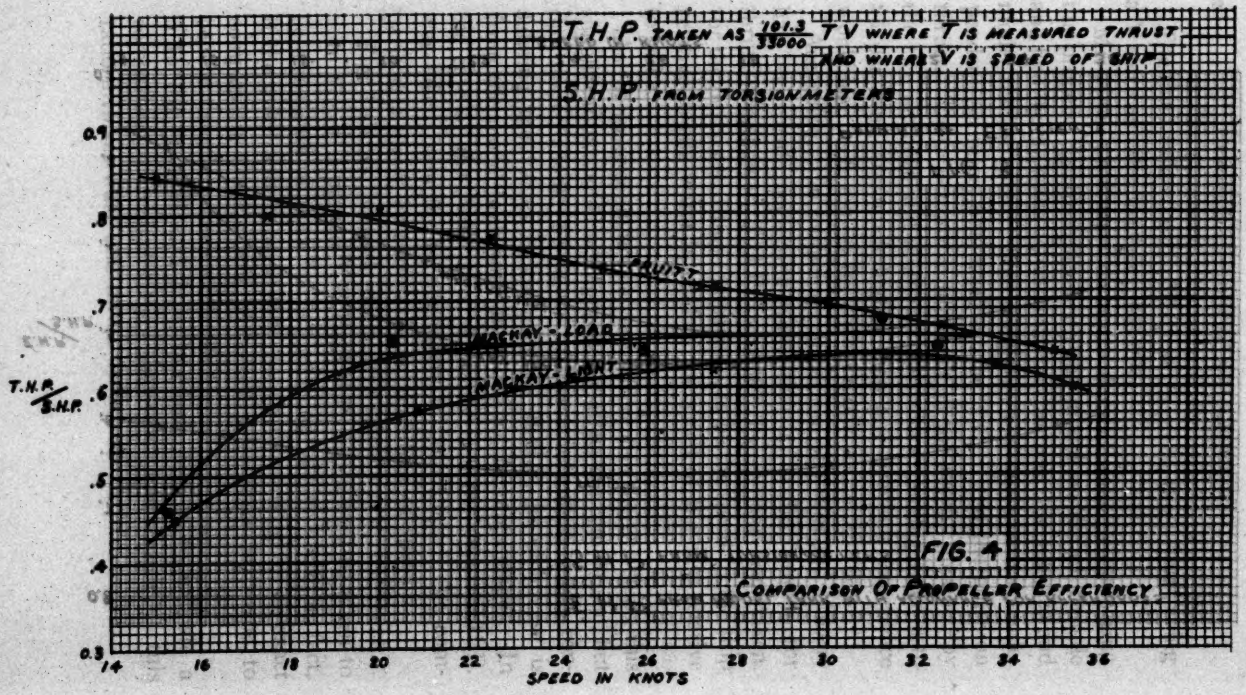


FIG. 4

COMPARISON OF PROPELLER EFFICIENCY

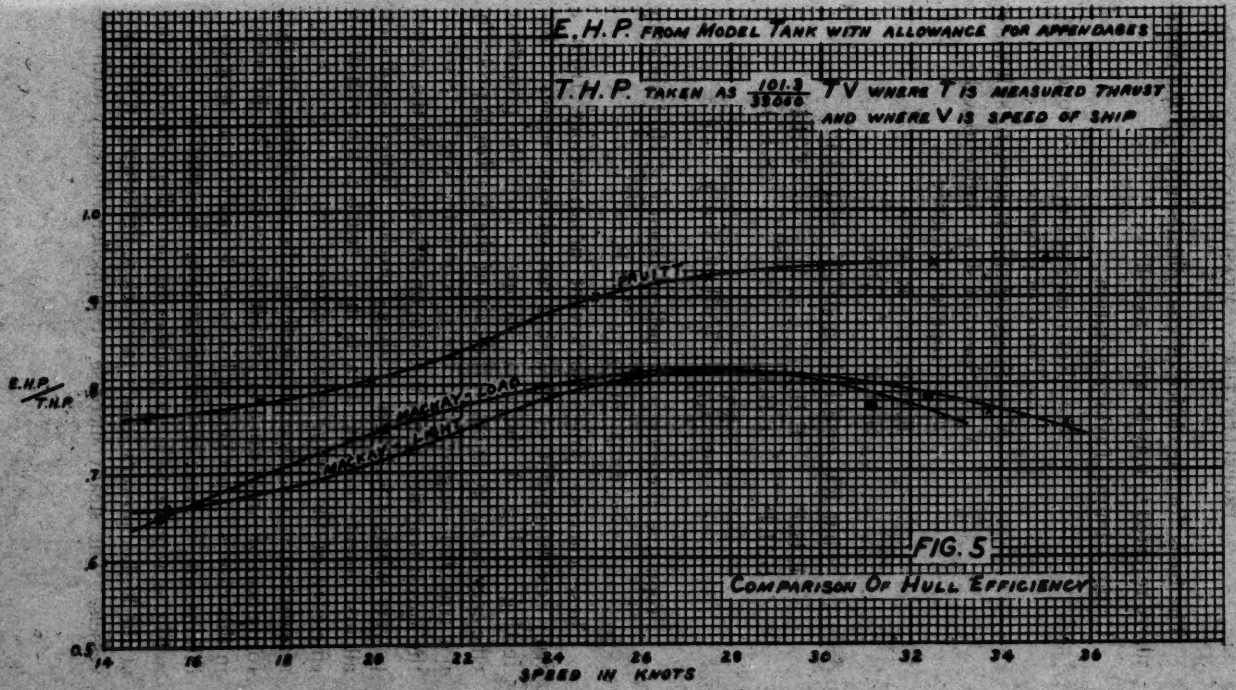


FIG. 5
COMPARISON OF HULL EFFICIENCY

The basis for the choice of co-ordinates used in the curves above referred to may be given as follows:

Using Admiral Taylor's notation:

If

R = Resistance of ship (tow rope)
T = Thrust of propeller
V = Speed of ship relative to still water
V_1 = Speed of ship relative to wake
e = Propeller efficiency

$$\frac{\text{E.H.P.}}{\text{S.H.P.}} = e\,\frac{R}{T}\,\frac{V}{V_1}$$

but $RV = \text{E.H.P.}$ and $TV_1 = \text{T.H.P.}$

$$\therefore \frac{\text{E.H.P.}}{\text{S.H.P.}} = e\,\frac{\text{E.H.P.}}{\text{T.H.P.}} \qquad \text{also } e = \frac{\text{T.H.P.}}{\text{S.H.P.}}$$

$$\text{and } \frac{RV}{TV_1} = \text{Hull efficiency}$$

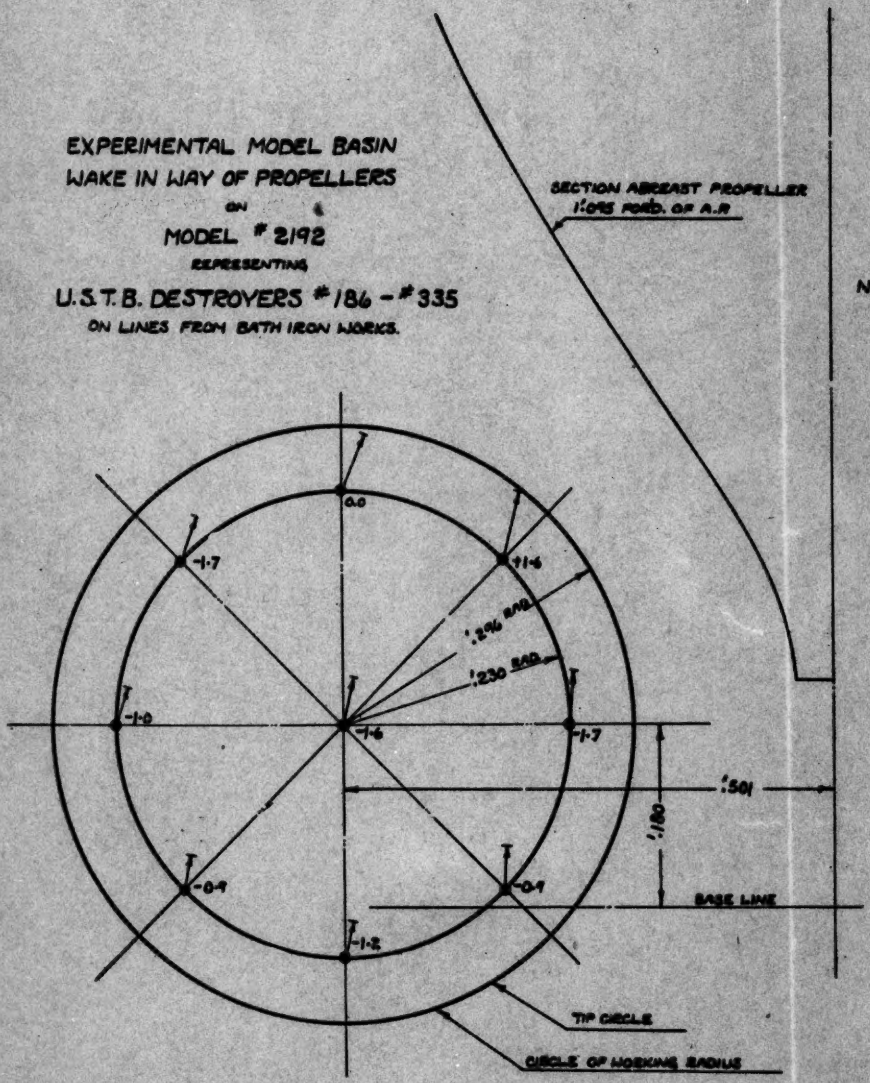
$\therefore$ Propulsive efficiency = Propeller efficiency $\times$ Hull efficiency

Note that in above $\text{T.H.P.} = TV_1$

These curves, though somewhat unsatisfactory from a truly scientific point of view, are not without interest. Referring to Fig. 3, the rapid falling off of the propulsive efficiency below 20 knots in the case of the *Mackay* is probably not true and suggests that the torsionmeter was in error at the lowest readings, a not uncommon difficulty. At the higher powers where the larger readings produce better results on all instruments and the data is more reliable, the propulsive efficiency of the *Pruitt* greatly exceeds that of the *Mackay*, a feature that has often been noticed when comparing the performances of American and British destroyers. The difference in this case is 20 per cent at load and 84 per cent in light conditions, and is great enough to invite an explanation.

Fig. 4 represents a comparison of propeller efficiency and at the higher powers shows fairly close agreement in the two vessels. The steady increase of propeller efficiency with re-

EXPERIMENTAL MODEL BASIN
WAKE IN WAY OF PROPELLERS
ON
MODEL # 2192
REPRESENTING
U.S.T.B. DESTROYERS #184 - #335
ON LINES FROM BATH IRON WORKS.



SPEED RATIO OF SHIP TO MODEL 3.94
TRIM EVEN KEEL
LINEAR RATIO OF SHIP TO MODEL 15.5
SPEED OF MODEL 7.61 KNOTS
SPEED OF SHIP 30.00 KNOTS
DISPLACEMENT 714.00 LBS.
SCALE = FULL SIZE OF MODEL

NOTE :-

ARROWS ARE VECTORS SHOWING TRANSVERSE COMPONENTS OF WAKE TO A SCALE OF $\frac{1}{2}$" = 10 % OF SPEED OF MODEL.

NUMBERS INDICATE FORE AND AFT COMPONENTS OF WAKE IN PERCENT OF SPEED OF MODEL.

CIRCLES AT EXPERIMENTAL POSITIONS ARE ARBITRARY LIMITS FOR THE SAME.

EXPERIMENTAL POSITIONS WERE TAKEN ON A CIRCLE OF $\frac{7}{8}$ THE DIAM. OF THE PROPELLER, AS THIS IS TAKEN TO BE THE WORKING RADIUS OF THE PROPELLER BLADES.

FIG. 6

duced speed in the case of the *Pruitt* suggests that the thrust readings at low powers are too high, since this feature was not present in Fig. 3, and this suggestion is supported by the somewhat unnatural falling off of the hull efficiency at low speeds.

Fig. 5, comparing hull efficiency, shows a marked advantage in favor of the *Pruitt*.

Bearing in mind that all measurements are subject to much greater errors at low speed than at full speed, the comparison seems to indicate that the advantages gained by the *Pruitt* were due to the factors affecting wake and thrust deduction, namely, the design of the hull and the appendages aft, rather than to the design of the propeller.

COMPARISON WITH MODEL PROPELLER EXPERIMENTS.

An attempt was made to find the relation between the actual performance of the *Pruitt's* propellers and that of a model propeller of similar proportions. This would not be very difficult were it not for two factors, wake coefficient and virtual pitch.

The wake of the *Pruitt's* model was investigated in the Model Basin at a speed corresponding to 30 knots, using Pitot tubes distributed over the area occupied by the propeller. By courtesy of the Bureau of Construction and Repair the results are shown on Fig. 6, from which it will be seen that for the speed of 30 knots there is practically no wake, the average of the spots being something less than 1 per cent negative. For speeds in the neighborhood of 30 knots, therefore, no great error would be involved if the wake be taken as zero.

The face pitch of the propeller was accurately known to be 10.17 feet, and the virtual pitch, or pitch of zero thrust (equal to speed of advance divided by revolutions at no thrust) must be obtained by inspection.

ADMIRAL TAYLOR'S EXPERIMENTS.

In using Admiral Taylor's model experiments, his original paper published in 1904 before the Society of Naval Architects and Marine Engineers was used and in the form that the results are presented it is convenient to make the comparison on a basis of S.H.P.

The formula is:

$$S.H.P. = .00312\, n\, S\, D^2\, V^2 \quad \ldots\ldots\ldots\ldots\ldots (1)$$

where n = Number of blades.

V = Speed of advance of propeller relative to the water in which it works.

S = A coefficient found from the model experiments and depending upon slip as the only variable.

Wake and virtual pitch affect both V and S in the above formula and by inspection it was found that the best agreement between the S.H.P. calculated by the formula and that actually measured by the torsionmeters was obtained by taking the wake as zero and the virtual pitch equal to 10.77 feet or 1.06 times the face pitch. The results thus obtained are here given, S.H.P. being for one screw:

Knots	15	20	25	30	35
S.H.P. by model	625	1580	4130	8720	20450
S.H.P. torsionmeter	640	1700	4300	8575	13725

There is reasonably close agreement except at 35 knots, where there is a wide discrepancy. Any attempt to adjust this by using different values of virtual pitch and wake were unsuccessful. It is possible that the wake may have another value at speeds above 30 knots, but it would hardly be likely to reach 16 per cent negative, the value necessary to bring the model S.H.P. down to 13725, and of course another value of the virtual pitch selected to adjust the S.H.P. at 35 knots would affect the agreement at other speeds.

A possible explanation of the discrepancy at 35 knots is that the model values of S were extrapolated in this region, or there may be partial cavitation in the actual screw which would not be present in the model, and which, while perhaps too slight to seriously affect efficiency, would permit of an increase in revolutions sufficient to give an augmented slip that would quickly affect S.H.P. through the steep S curve of the model experiments.

A further suggestion that the S.H.P. by the torsionmeters may be wrong is untenable, because this is checked by the steam consumption which in turn was checked by the fuel oil consumption, both of which were measured on the *Pruitt* and would have indicated any error in S.H.P. exceeding about 5 per cent at full power.

The propeller efficiency as already shown is given by T.H.P./S.H.P. where T.H.P. = .00307 T V, T being the measured thrust and V the speed of ship for zero wake.

Knots	15	20	25	30	35
Model propeller efficiency (Taylor)	.582	.59	.622	.635	.637
Actual propeller efficiency T.H.P./S.H.P.	.843	.803	.737	.696	.641

Although in fair agreement at the higher speeds, this comparison bears out a previous indication that the thrust measurements at lower powers gave results much too high.

R. E. FROUDE'S EXPERIMENTS.

These experiments were the subject of a paper before the Institution of Naval Architects in 1908 and a comparison between these and Admiral Taylor's experiments, made by T. B. Abell in 1910, showed the results of the independent investigations to be in fair agreement.

Froude's analysis is given in a form convenient for comparing his model results with the actual performance of the *Pruitt* on a basis of thrust. The method is as follows:

$$\text{Thrust} = a\, D^4\, R^2\, S \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (2)$$

where D = diameter in feet = 9.17 in case of *Pruitt*
R = revolutions per minute in 100's
S = slip
a = experimental coefficient.

He found the coefficient "a" to be a function of pitch ratio, disc area ratio and slip given by the relation

$$a = B\, p\, (p + 21) \times 1.02\, (1 - .08\, S) \ldots\ldots\ldots (3)$$

in which p is the pitch ratio and B a factor depending upon the disc area ratio only.

Substituting in (3) the values of p and B for the *Pruitt's* propeller and neglecting the term $(1 - .08\, S)$ which has practically no effect within our range of slip, we find $a = 3.27$.

On substituting for a and D in (2) the equation becomes:

$$\text{Thrust} = 2.33\, R^2\, S$$

R being revolutions per minute and S real slip.

The value of p to make the calculated thrust equal to the measured thrust at 30 knots (the speed at which the model tank experiments showed the wake to be about zero) was found by inspection to be 1.198, and this value was used in above. This means that the virtual pitch in this case must be taken at 1.08 times the face pitch.

The following table gives the computation on the basis outlined above, the wake being taken at zero for all speeds, and, as before, the thrust is for one propeller.

FROUDE ANALYSIS—NO WAKE.

Speed in knots	15	20	25	30	35
Revolutions	161	217	284.5	355	442.5
App. slip computed on face pitch	.0725	.08	.125	.157	.21
Real slip (virtual pitch = 1.08 face pitch)	.141	.150	.190	.221	.271

Thrust (model) =					
2.33 R^2S	8500	16500	35800	64800	123800
Thrust measured	10950	22250	41350	64800	81950
Efficiency of model propeller	.637	.647	.670	.675	.667
Efficiency of actual propeller = T.H.P./S.H.P.	.843	.803	.737	.696	.641

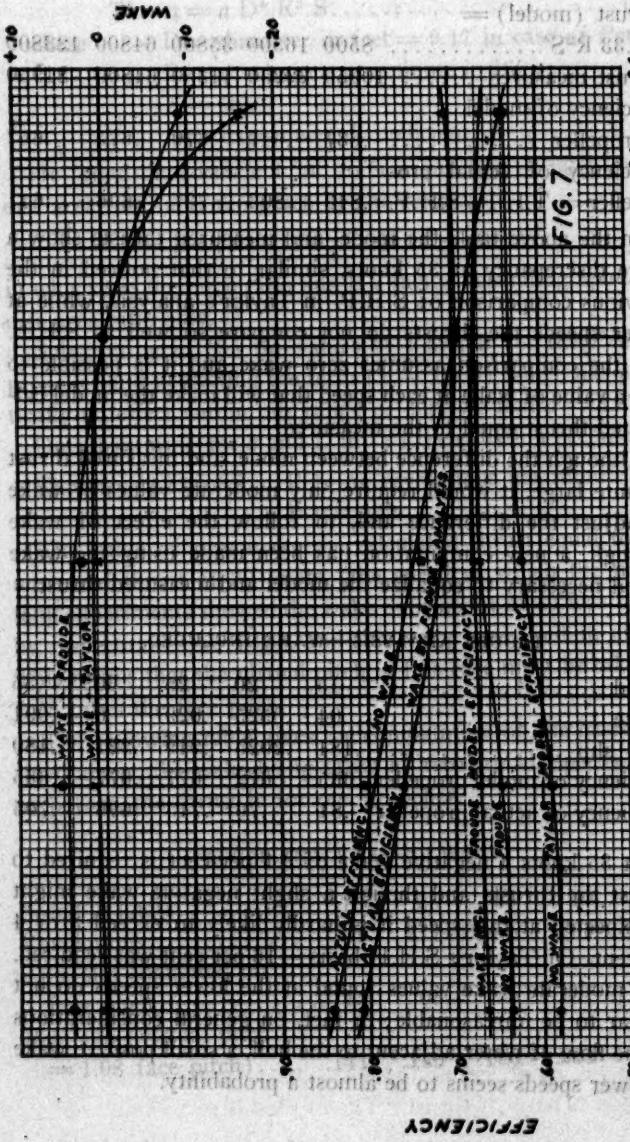
In the above table the model and measured thrusts show a large discrepancy at 35 knots, similar to that noticed in the previous comparison of S.H.P. in Taylor's analysis, while at lower speeds the thrusts do not compare so well as S.H.P. But the comparison assumes zero wake and it is possible to find a value of wake at each speed that will make the model calculated thrust equal to the measured.

Although the difference between model and measured thrust is quite large, it would require only moderate values of wake to adjust the difference and, to follow the effect of wake through, a new computation has been made using the wake found necessary to equalize the model and measured thrust.

FROUDE ANALYSIS—WAKE INCLUDED.

Speed	15	20	25	30	35
Wake	.04	.0525	.029	0	—.091
Real slip	.181	.2025	.219	.221	.180
Efficiency of model propeller	.665	.672	.677	.677	.665
Efficiency of actual propeller	.81	.76	.715	.696	.705

At 35 knots a negative wake of 9.1 per cent is required to adjust the thrusts and though a slight negative wake might be expected at this speed it is hardly likely to exceed 3 or 4 per cent. On Taylor's S.H.P. basis —16 per cent was required. The moderate wake values found at the lower speeds do not appear to be unreasonable; in fact, on general considerations of the flow of water adjacent to the hull, a small positive wake at lower speeds seems to be almost a probability.


WANE
+20
0
-20
-20
FIG. 7
SPEED IN KNOTS
EFFICIENCY
SCALE OF REVOLUTIONS
SCALE OF THRUST
SCALE OF EFFICIENCY
SCALE OF SLIP
SCALE OF TORQUE
SCALE OF POWER

On Fig. 7, curves of wake and a propeller efficiency as deduced from Taylor's and Froude's analysis are shown plotted to a base of speed.

CONCLUSIONS.

At 35 knots the thrust and S.H.P. were much less than the model experiments would lead us to expect from the propellers at the revolutions and speed recorded, and from this one is led to the belief that partial cavitation existed in the actual screws, although this was apparently not sufficient to appreciably affect efficiency.

The efficiency of the actual screws seems to be about equal to that of the model screw run in open and undisturbed water. This conclusion runs contrary to the general belief that a deduction in efficiency must be expected due to the turbulent character of the water in which actual propellers work. Mr. Holt thought that this allowance was 15 per cent in the case of the *Mackay*.

The analysis emphasizes the necessity for more knowledge regarding wake and the ratio that the virtual pitch bears to the nominal or face pitch of propellers of various proportions, without which the available model propeller results cannot be fully utilized. In the case of the *Pruitt* the results seem to indicate that the wake must be nearly zero; probably a small positive wake exists at lower speeds and a small negative wake at the maximum speed. The value of the ratio of virtual to face pitch is about 1.07. The model tank can best provide these data which can be conveniently checked by actual thrust measurements.

The thrust measurements on the *Pruitt* were undoubtedly too high at low speeds. At the time the readings were taken the operators were very dubious about results at 15 and 20 knots. The pressures recorded at these speeds are small and for that reason alone are susceptible to greater errors, but it was also noticed that, although conditions in the engine room

might be absolutely steady, the effect of the helm in the ordinary course of steering was quite noticeable on the gauge. The slight wallowing at low speeds in even a moderate sea can thus have considerable effect, although nearly absent at higher speeds.

Facility in the operation of the apparatus would be greatly improved by the use of an ordinary steam engine indicator connected to the oil space and used in conjunction with a reservoir of oil at a convenient pressure. A large number of instantaneous readings would then be obtainable in the convenient form of an automatic record, and errors due to helm and variable immersion could be eliminated.

That large thrusts can be conveniently and accurately measured was demonstrated, and, in the case of geared turbine installations with Kingsbury thrusts on the forward end of the gear box, thrust is even easier and more accurate to measure than I.H.P. or S.H.P.

In addition to propeller and hull efficiency, other very desirable data is available from thrust meter records, for instance increase of resistance due to head wind and weather, fouling of ship's bottom, and the variation of thrust in backing, stopping and turning.

The writer acknowledges his indebtedness to the Bath Iron Works, Ltd., the builders of the *Pruitt*, for permission to publish this paper, and for the data on which it is based.

PILOTING VESSELS BY ELECTRICALLY ENERGIZED CABLES.*

By A. Crossley, Associate.

Piloting vessels by means of cables energized by audio frequency currents is the result of the work of many scientists, inventors and engineers, each having contributed some part to the general scheme, whereby it is possible at this date, to successfully apply the system for piloting vessels through treacherous channels during inclement weather.

Before entering into a discussion on the general details of this system, the writer takes the liberty of giving a brief history of the evolution of the system. This history is the result of a thorough investigation of the prior art. The writer does not wish to be quoted officially as to the authenticity of statements made, should it be proven later that there are other inventors who claim priority on the hereinafter stated inventions. The writer therefore does not desire to cover the system from a patent standpoint, but only desires to bring to the attention of those interested the technical details, and also the possibilities of this new aid to navigation.

GENERAL HISTORY.

A resumé of the past history of the evolution of the Audio Piloting System may be briefly stated as follows:

Michael Faraday discovered, in 1831, the principle of electro magnetic induction.

Alexander Graham Bell's invention of the telephone receiver in 1876.

*Paper also to be presented before the Institute of Radio Engineers.

A. R. Sennett's British patent, No. 13,415, of 1892, which discloses means of maintaining communication with light vessels by use of a submerged cable from which signals were received on board lightship by means of a multiple turn loop and telephone receivers.

Charles A. Stevenson's British Patent, No. 5,498, of 1892, which discloses means for navigating vessels over electrically energized cables and which shows the use of conduction principle at the receiving end in connection with sensitive galvanometer for recording purposes.

Robert B. Owens' American patent, No. 736,432, of 1903, which shows a combination of the Stevenson's cable and Sennett's receiving apparatus in conjunction with special method of tilting receiving coils with respect to cable.

Alexander Chessin's British patent, No, 21,610, of 1912, which shows an improvement on Owens' patent with reference to a means of mechanically tilting receiving coils.

July, 1916, issue of "Motor Boating," and also the August, 1916, issue of "Popular Mechanics," in which Earl C. Hanson discloses information on application of vacuum tube amplifier to audio frequency circuits among which is the proposed use of such a system for piloting vessels through fogs.

Earl C. Hanson's American patent, No. 1,315,197, of 1919, which discloses application of vacuum tube amplifier to audio frequency receiving circuits.

Further work on the application of the amplifier to the Audio Piloting System has been, it is understood, undertaken by the English and German navies, but full details on this work are not known to the writer.

Expert Radio Aid R. H. Marriott, of the Puget Sound Navy Yard, conducted a series of interesting experiments at that place using the same circuits that Hanson had previously employed. Marriott brought to the attention of the Navy Department the results of his tests, and partly through his efforts the

Navy Department decided to make a thorough investigation of the merits of the system.

Having outlined the previous work attempted, we will now devote our attention to the experimental work undertaken by the Bureau of Engineering, Navy Department, to determine the feasibility of developing and applying this system for Naval use.

NEW LONDON EXPERIMENTS.

Upon receipt of orders, the writer proceeded to the Submarine Base, New London, Connecticut, on October 18, 1919. to conduct preliminary experiments. Arrangements were made whereby a sixty-foot sailing launch was placed at my disposal for experiments. A 5,250-foot length of 9,000 circular mil single conductor (Navy Standard) rubber insulated cable was submerged in the channel of the Thames River from the Submarine Base to a point approximately 5,000 feet down the river. The extreme end of this cable was grounded to a six-foot length of 1½-inch galvanized iron pipe while the shore end of the cable was connected through a lamp bank of eight 32-candlepower carbon lamps to a ½-KW., 500-cycle generator on board the submarine N-4, and grounded to the hull of the submarine. The cable was anchored at 500-foot intervals along the course, but due to the difficulty experienced in handling the sailing launch, this cable was not laid in a straight course.

The receiving equipment consisted of two coils wound on a wooden frame four feet square, each coil having 240 turns of No. 20 s.c.c. copper wire, a double-pole double-throw switch and a type SE 1,600 Navy two-stage audio frequency amplifier, to which was connected standard telephone receivers. The double-pole double-throw switch made it possible to connect either coil to the amplifier, and thereby compare the relative received signal strength of the respective coils. A .0025 mf. variable air condenser was connected in series with

the input terminals of the amplifier as shown in Fig. 1, with a view to tuning the circuit to resonance; but, due to the low

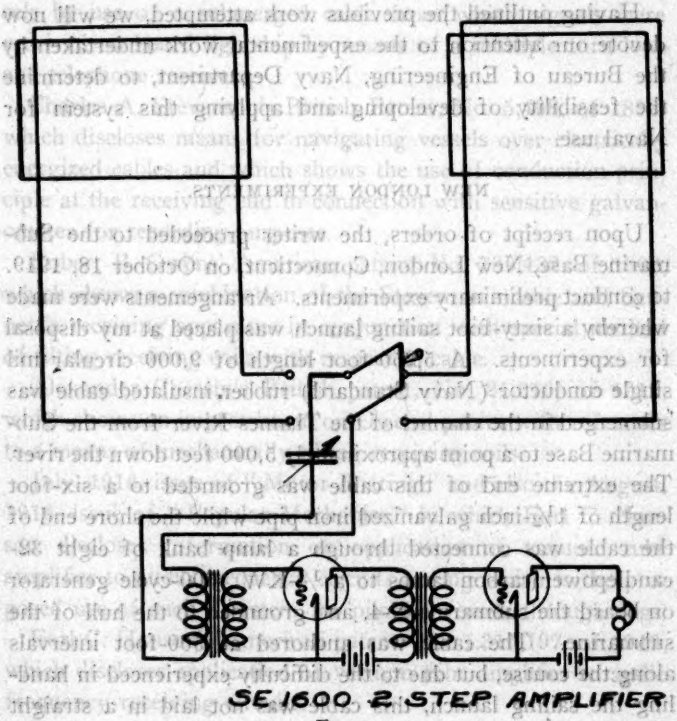


Fig. 1.

inductance of the coil, this was not possible, and the condenser was removed from the circuit.

A current flow of 2.3 ampères at 170 volts was supplied to the submerged cable. The input watts as recorded on a wattmeter, indicated that the input in the cable was 145 watts, or a power factor of .37. The poor power factor obtained was due to the fact that it was not convenient to obtain direct leads from the generator without totally dismantling the panel of the radio set of which the generator was a part, and

therefore resort was made to connecting the primary of the transformer in series with the cable, thus creating a predominance of inductance in the circuit with the resultant lagging power factor. The receiving apparatus was installed on the sailing launch, one coil on one side and the other coil on the opposite side of the launch, above the water line. The launch was then made to follow a zigzag course over the cable to note the relative signal strength as received from the cable by the respective coils. It was noted that there was no shielding effect between the two coils and resort was made to the use of one coil. Using the single coil in the vertical position, it was possible to tell within a distance of ten feet the exact location of the cable. The signal received from the cable was of an audibility in excess of 10,000 when the launch was directly over the cable, and of such intensity as to make it uncomfortable to retain telephone receivers on the observer's ears, while when moving away from the cable the signal strength rapidly diminished until it became inaudible at a distance of 600 feet from the cable.

The next experiment consisted of turning the coil at different angles to the vertical plane to note whether a pronounced maximum or minimum signal could be obtained from the cable. It was noted, as expected, that a maximum signal was obtained when the plane of the coil was in line with the cable, and a minimum signal was obtained when the coil was at right angles to the plane of the cable.

Having proved that the single coil could be used to advantage on wooden vessels, arrangements were made with the Submarine Base, whereby the Submarine G-1 was detailed for work to prove whether or not a shielding effect was present when a steel hull was interposed between the coils.

The results of experiments conducted on the G-1 proved that there was a shielding effect between the coils and by use of this effect the Commander of the G-1 could easily note on which side of the submarine the cable was situated. The

stronger signal was always picked up by the coil nearer the cable, while minimum signal was received by the coil which was further from the cable. The shielding effect was not as pronounced as it should have been, but this was probably due to the poor shielding effect of the sloping sides and small conning tower of the G-1.

On going down the river and passing over end of cable, it was immediately noted that the submarine had passed over the cable by the sudden decrease in signal strength. This decrease in signal strength was much more pronounced than the decrease noted when passing over and away from cable on a course at right angles to cable. No difference was noted in signal strength when the submarine was at different points along cable, showing that the attenuation effect along the cable could be neglected from a practical operating standpoint.

Having concluded necessary preliminary experiments at New London, the writer returned to Washington and recommended to the Bureau of Engineering that this system be further investigated wherein resort could be made to the use of a vessel having steel hull with a high superstructure, and that experiments be conducted under conditions which would be representative of its future practical application. After consideration of the recommendations, the Bureau of Engineering decided that New York Harbor would be an ideal place to complete the final experiments.

NEW YORK EXPERIMENTS.

Immediately after the decision to conduct further experiments at New York, the writer proceeded to the Navy Yard, New York, and conferred with Commander R. F. McConnell, and made arrangements for the future experimental work. The New York Yard then obtained the services of the Mine Planter *Ord* and laid a cable from the Bay Ridge Naval Station to a point approximately 700 feet from the Ambrose Channel Lightvessel. The cable referred to above consisted

of 2,000 feet of 9,000 c.m. leaded and armored cable, 2,000 feet of 9,000 c.m. lead-covered cable and 83,000 feet of 9,000 c.m. rubber-insulated Navy Standard cable. This cable was anchored at 500-foot intervals with concrete anchors of similar design to anchor shown in Fig. 2, and was laid as close as

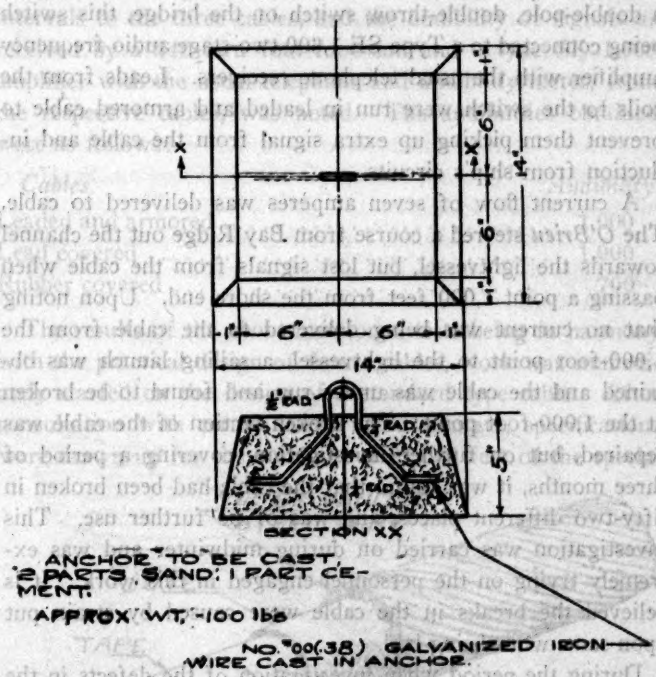


FIG. 2.

possible to the center of Ambrose Channel. The extreme end of the cable was grounded to a two-foot square copper plate while the shore end was connected through an inductance to a relay key which was actuated by an Omnigraph transmitter, and from this key connection was made to one terminal of a ½-KW., 500-cycle generator; the other terminal of the generator was grounded to a copper plate two feet square, which was submerged in the bay.

The destroyer *O'Brien* was detailed to assist the New York Yard in this work and was equipped with two collector coils, four feet square, having 240 turns of No. 20 d.c.c. wire, one coil suspended over one side of the vessel and the other coil over the opposite side. Leads from these coils were led to a double-pole, double-throw switch on the bridge, this switch being connected to a Type SE 1,600 two-stage audio frequency amplifier with the usual telephone receivers. Leads from the coils to the switch were run in leaded and armored cable to prevent them picking up extra signal from the cable and induction from ship's circuits.

A current flow of seven ampères was delivered to cable. The *O'Brien* steered a course from Bay Ridge out the channel towards the lightvessel, but lost signals from the cable when passing a point 1,000 feet from the shore end. Upon noting that no current was being delivered to the cable from the 1,000-foot point to the lightvessel, a sailing launch was obtained and the cable was under run and found to be broken at the 1,000-foot point. The broken section of the cable was repaired, but on further investigation, covering a period of three months, it was found that the cable had been broken in fifty-two different places, and was of no further use. This investigation was carried on during midwinter and was extremely trying on the personnel engaged in this work. It is believed the breaks in the cable were caused by strain put upon same while being laid.

During the period when investigation of the defects in the cable was being conducted, the New York Yard was requested to try out different types of cable to note which type would be best suited for this system. This was the original idea when the three different sections of cable were incorporated in the first cable, but as it was not possible to obtain this information due to the defective nature of this cable, experiments were conducted in the Navy Yard to solve this problem. Three cables one of 9,000 c.m. leaded and armored, one lead covered 9,000 c.m. and one rubber insulated 9,000 c.m. cable, each

having a length of 150 feet, were submerged to the same depth in the East River near Pier H, in the Navy Yard. Each cable was grounded at one end to a copper plate and the other end was connected to a ½-KW., 500-cycle generator. A current flow of one and one-half ampères was supplied at different intervals to the three cables, and the audibility of signals as received by a 240-turn four-foot coil and a type SE 1,600 amplifier with the usual telephone and audibility meter, from the respective cables, was noted. The audibilities obtained were as follows:

Cables.	*Audibility.*
Leaded and armored	1,000
Lead covered	1,000
Rubber covered	700

The results of the above-mentioned tests being in harmony with the practical design of future cables, work was immediately started on the design of a standard type cable. After consultation with certain cable manufacturers, specifications were drawn up for the purchase of 87,000 feet of the special

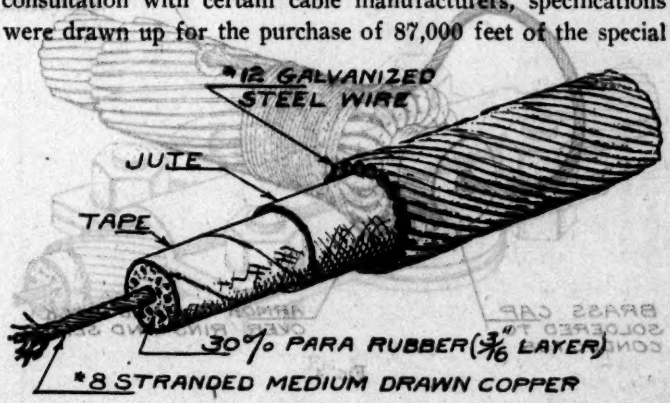


FIG. 3.

cable shown in Fig. 3. This cable consists of seven strands No. 16 tinned copper wire, layer of 30 per cent Para rubber 3/16-inch thick, one layer of cambric, layer of jute and an

armor of No. 12 galvanized steel wire. This cable was manufactured by the Simplex Wire and Cable Co. of Boston. The cable was delivered by motor truck to the Navy Yard, Boston, in four reels. The Coast Guard Cutter *Pequot* was detailed to lay the new cable in Ambrose Channel and was at the Navy Yard, Boston, to receive this cable. Upon arrival of the four reels at the Navy Yard, representatives of the cable company made the necessary cable splices so that it was stowed on board the *Pequot* in one continuous length. This procedure was of great help, as it eliminated the extra work and delay which would have been encountered should splices be made during cable-laying operations.

While the cable was being put on board the *Pequot* a special connection was made to the end of the cable. Fig. 4 shows a

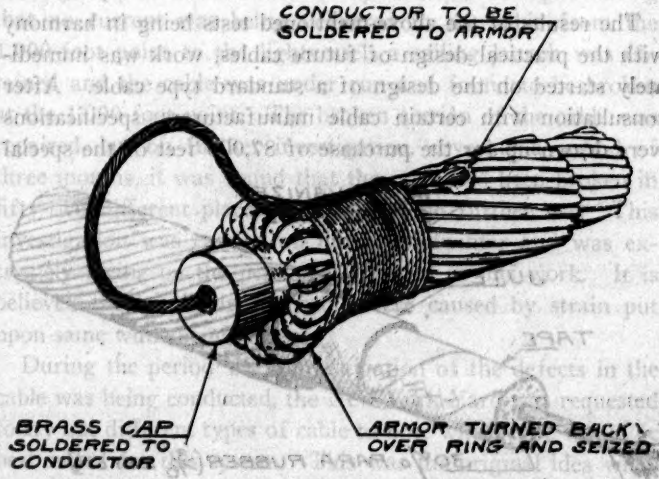


FIG. 4.

sketch of this connection. The connection made between the internal conductor and the armor was for the purpose of eliminating the grounding plate and the attendant mechanical and electrical difficulties encountered when using this plate.

It was figured theoretically that the low resistance offered by the large surface of the grounding plate (4 square feet) to the return current through salt water between the plate and the armor was approximately equal to resistance of the short length of the internal conductor (6 inches) as shown in Fig. 4. The brass cap covering the rubber insulation at end of wire was used to keep the salt water from seeping back into the cable between rubber insulation and the wire.

The *Pequot* arrived at New York July 31, 1920. Three days were spent in underrunning the Western Union, Army and Police cables which cross the Narrows between Forts Hamilton and Wadsworth, after which operation the "Piloting" cable was placed under these cables. This was accomplished by hoisting up cables one by one and passing a 60-foot sailing launch with 2,000 feet of cable on board under the cables and then allowing cables to be released and sink to bottom. After this operation the sailing launch carried the remainder of the cable ashore and placed 500 feet of cable on the dock at Fort Lafayette.

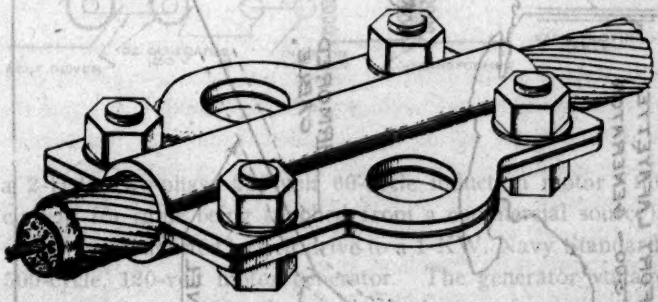


FIG. 5.

When the shore end of the cable was secure, the *Pequot* headed for the middle of the channel and on reaching this point on the course a cable clamp (see Fig. 5) was placed on the cable, and by means of shackles and 5/16-inch chains, an

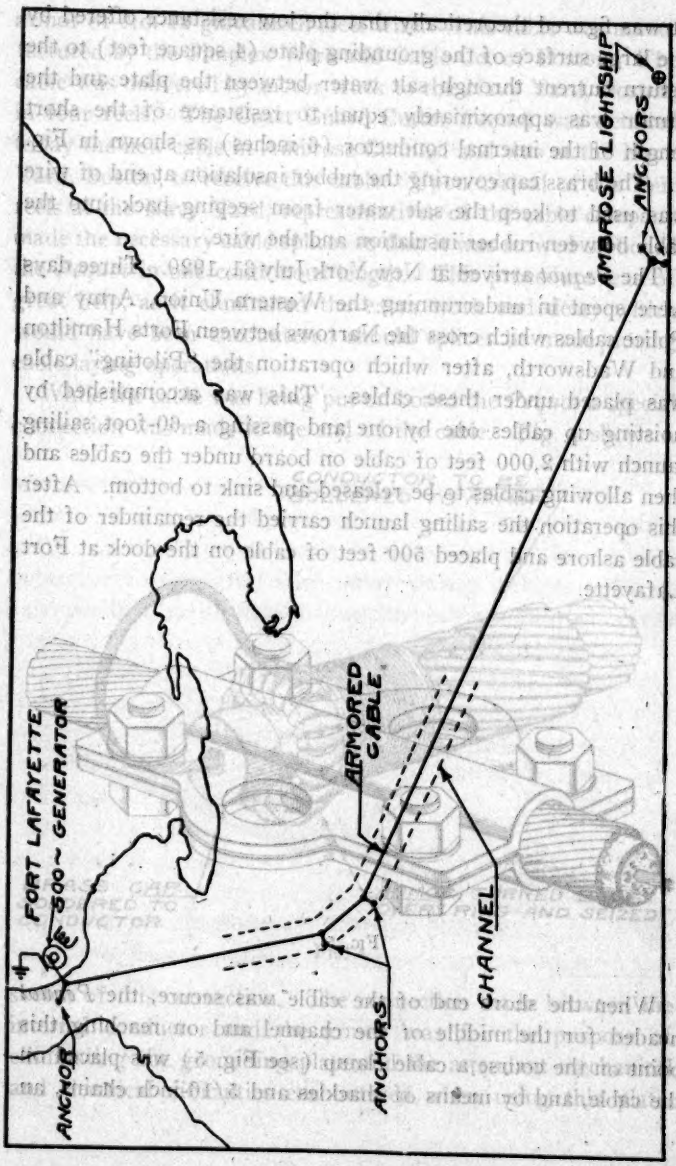

AMBROSE LIGHTSHIP
ANCHORS
ARMORED CABLE
FORT LAFAYETTE
300~ GENERATOR
ANCHOR
ANCHORS
CHANNEL

FIG. 6.

anchor was made fast to cable and lowered over the side. From this point the *Pequot* followed as close to the middle of the channel as was possible on her way out of the Ambrose Channel, passing to the left of the entrance buoy and completing the work of laying the cable when at a point 1,900 yards from the Ambrose Lightvessel. The cable was anchored at the two turns and also at the end as shown in Fig. 6. This cable, approximately 16 miles in length, was laid in a period of six hours. Upon completion of the cable laying, August 6, 1920, test was made to ascertain whether the cable was in good condition, the result of which test showed that it was possible to obtain a current flow of 3½ ampères when using 400 volts supplied by 1-KW., 500-cycle generator at Fort Lafayette. Upon noting that the cable was apparently in good condition a power unit was installed at Fort Lafayette for this cable. This unit shown in the schematic diagram, Fig. 7, consisted of

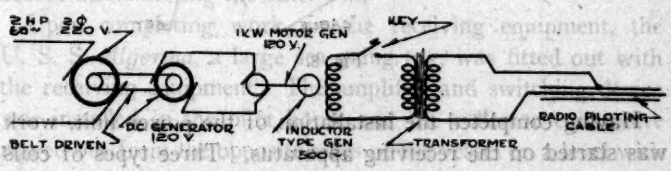


FIG. 7.

a 2-H.P. two-phase 220-volt 60-cycle induction motor (the current for same being supplied from a commercial source), which was connected by belt drive to a 1-KW. Navy Standard, 500-cycle, 120-volt motor generator. The generator voltage was stepped up to 400 volts by means of a transformer and the output of the transformer supplied direct to the cable. An Omnigraph transmitter connected to a Navy Standard relay key was used to send distinguishing signals into the cable. After improving the receiving equipment, it was found that

the step-up transformer could be dispensed with, thereby simplifying to a certain extent the power unit. (See Fig. 8.)

The cable before being laid was found to have an impedance of 50.2 ohms at 500 cycles and a direct current resistance of 50 ohms. The capacity of the cable measured at 800 cycles was 5.7 mfs. After laying the cable, it was found that the impedance of the cable circuit was 123 ohms at 500 cycles, and that there was a lagging power factor, which when measured was found to be .80. The inductance (calculated) of the cable was 38.28 milli-henries.

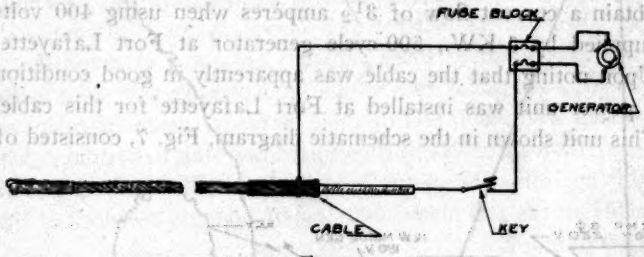


FIG. 8.

Having completed the installation of the power unit, work was started on the receiving apparatus. Three types of coils were constructed, one having 400 turns, one 600 turns, and the other 800 turns, wound on a wooden frame four feet square. These coils, with the exception of the 400-turn coil, were wound with No. 20, B. & S. gauge, d.c.c. copper wire. The 400-turn coil was wound with No. 24 wire, which size wire gave considerable trouble as it stretched easily. It was therefore practically impossible to wind two coils with this wire and have them of the same resistance. It was imperative that both coils have the same resistance and therefore resort was made to rewinding the 400-turn coil with No. 20 wire which gave very satisfactory results. The coils were layer wound and impregnated with beeswax and boiled linseed oil.

The resistance of the coils are as follows:

Coil.	*Resistance.*
400 turn	64.3 ohms
600 turn	100.2 ohms
800 turn	130.9 ohms

Leads from each coil terminated in a watertight junction box on the coil frame, from which portable leads (leaded and armored duplex cable) were led to a double-pole double-throw switch and from there to a type SE 1,000' two-step audio frequency amplifier and the Navy telephone receivers.

The double-pole double-throw switch had to be redesigned so that it would at no time completely open the input circuit of the amplifier, as under such a condition the amplifier would howl and deafen the observer. A rotating switch was therefore designed which when rotated made contact with one coil before disconnecting the other coil.

Upon completing work on the receiving equipment, the U. S. S. *Algorma*, a large sea-going tug, was fitted out with the receiving equipment. The amplifier and switching device was installed in the pilot house and the collector coils were rigged out from the opposite sides of the vessel on a level with the upper deck and approximately amidships. In this condition the coils were about fifteen feet above the waterline. The coils were rigged on separate outriggers by use of which the coils could be swung clear of the ship's side in a vertical position to a maximum distance of six feet.

When all circuits (transmitting and receiving) were found to be in perfect condition, the *Algorma* steamed out the Ambrose Channel and conducted preliminary experiments on the system. During these experiments it was found that the steel hull of the *Algorma* had a very pronounced shielding effect on the respective coils, which was not noted during the New

London experiments. This shielding was such that it was possible to easily note when approaching the cable the exact location of the cable with respect to the vessel. When the vessel was directly over the cable a signal of the same audibility was received on each collector coil, while if the vessel moved to the right or left of the cable the coil nearer to the cable picked up the stronger signal, and the coil further from the cable picked up a signal of markedly less intensity.

Fig. 9 gives an idea of the relative audibilities received by

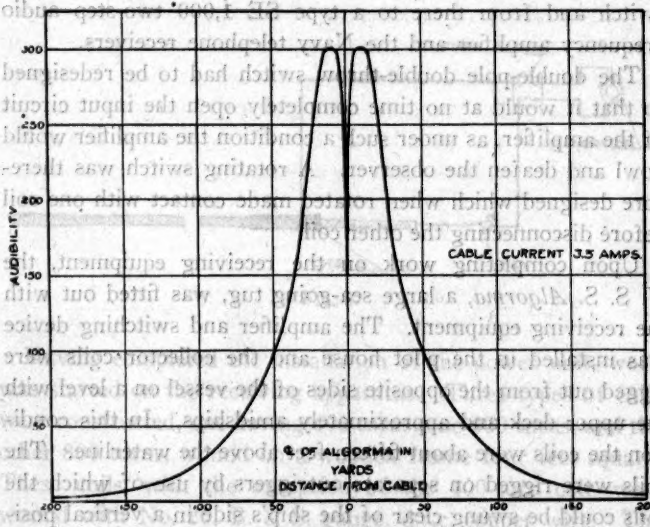


FIG. 9.

the nearer coil when vessel is at different distances from the cable. It can be seen from this graph that the shielding of the hull of the vessel is responsible for the big decrease in audibility when vessel is directly over cable, and the tremendous increase in audibility when the vessel is approximately 30 feet either side of the cable. Further explanation of this point may

be had by reference to Figures 10 and 11, which show the apparent effect of the electro-magnetic field on the respective collector coils.

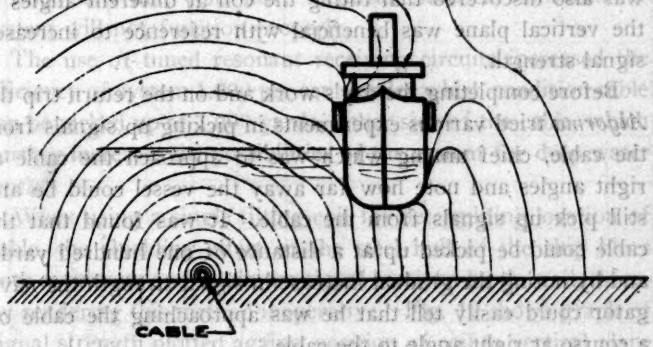


FIG. 10.

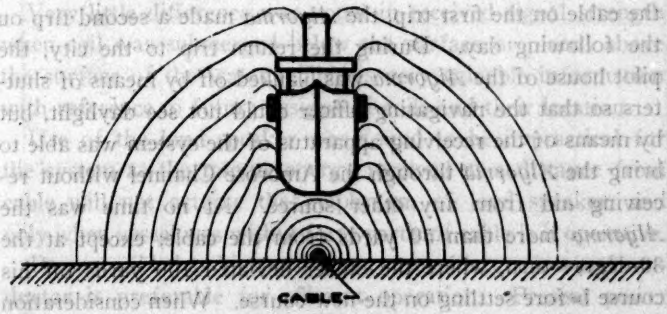


FIG. 11.

This pronounced shielding effect was found to be very helpful as by its use a vessel could keep the maximum signal in the port coil and steer a course which would always keep the vessel on the correct side of the channel.

After noting the shielding effect, experiments were conducted to ascertain the correct distance that the coils should be from the side of the vessel. The results of these experiments show that the coils should not, for a vessel of the *Algorma* type, be at a greater distance than three feet from the side of the vessel. Should this distance be exceeded the shielding effect of the hull would be annulled to a great extent. It was also discovered that tilting the coil at different angles to the vertical plane was beneficial with reference to increased signal strength.

Before completing the day's work and on the return trip the *Algorma* tried various experiments in picking up signals from the cable, chief among which was to approach the cable at right angles and note how far away the vessel could be and still pick up signals from the cable. It was found that the cable could be picked up at a distance of one hundred yards, and by noting the gradual increase in signal strength the navigator could easily tell that he was approaching the cable on a course at right angle to the cable.

Having determined certain major navigational features of the cable on the first trip, the *Algorma* made a second tirp on the following day. During the return trip to the city, the pilot house of the *Algorma* was blanked off by means of shutters so that the navigating officer could not see daylight, but by means of the receiving apparatus of the system was able to bring the *Algorma* through the Ambrose Channel without receiving aid from any other source. At no time was the *Algorma* more than 50 yards from the cable, except at the 30-degree turn, where the officer deviated 100 yards off his course before settling on the new course. When consideration is given to the fact that the navigating officer of the *Algorma* was a perfect stranger to the system, having only three hours' preliminary instruction, it is to be granted that this new aid to navigation has great possibilities.

During the succeeding tests on board the *Algorma*, much information was obtained, chief among which may be given as follows:—

A current flow of three ampères into cable is sufficient for all needs in water up to a depth of 200 feet.

Collector coil having 800 turns of wire gave twice the audibility of the 400-turn coil, which was to be expected.

Important that coils have identical electrical constants or system will not function properly.

The use of tuned resonant receiving circuits increased the efficiency of system 1,000 per cent, under which condition cable can be picked up at 1,000 yards, either side of the cable, which further increases the possibilities of the system for deep water work.

When using a single tilting coil for determining location of cable, the shielding effect of the steel hull of the ship blurs both maximum and minimum signal received by the coil, thereby rendering this method inoperative. Fig. 12 shows graph of signal strength plotted against position of coil to vertical plane when vessel was 30 yards from cable.

Very little difference was noted in received signal strength when coil was submerged below the surface or placed above the surface of the water. This is very valuable information with reference to future designs of coils and their location.

Use of the loud speaker was found to be impractical for the system as the minute energy received at a distance from cable will not actuate the diaphragm of loud speaker, and only when vessel was within forty yards of cable is it operative.

Present cable has high resistance; use of larger internal conductor is preferable for efficient operation. Present power costs ten cents per hour, while the use of larger internal conductor will reduce cost to five cents per hour. This may be accomplished by increase of 20 per cent over cost of present cable, by purchase of new type cable. Should old type cable be used for future installations, which require 20 miles or

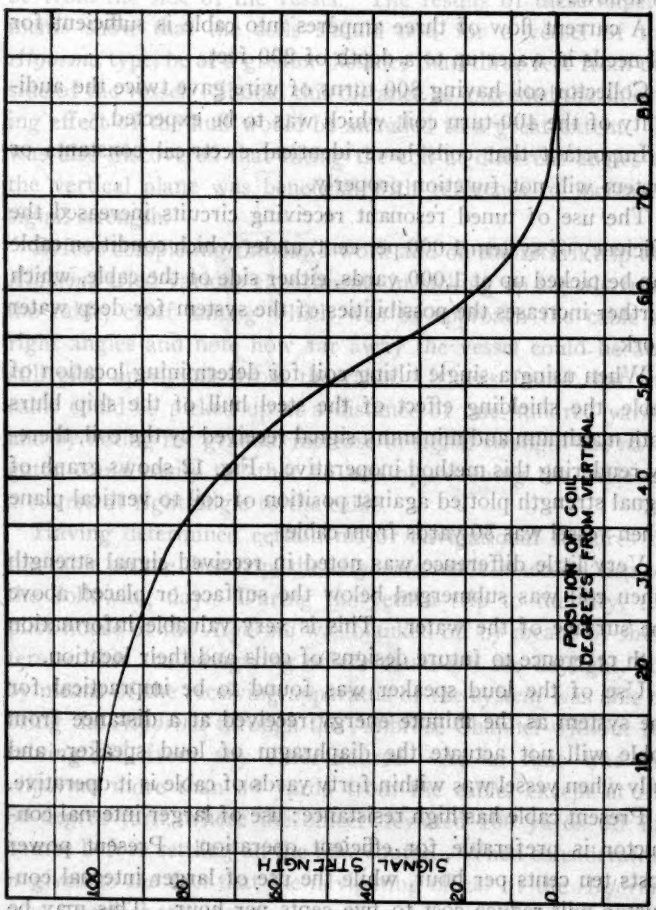

SIGNAL STRENGTH
100
80
60
40
20
0
POSITION OF COIL
DEGREES FROM VERTICAL
0
10
20
30
40
50
60
70
80

FIG. 12.

more of cable, a special 500-volt machine would have to be designed, while with the adoption of the larger cable the present Navy Standard motor generator units can be used.

The impedance of the return path through salt water and earth is theoretically many thousand times less than the impedance of the return path through the armor, and therefore the major part of the current will return through the water, thereby creating a resultant magnetic field. It is thought that the effect of the 500-cycle current through the cable has a restraining effect on the return current path through water, whereby it tends to converge this path close to the cable. This assertion is based on the fact that no stray current return paths were noted, which would have been the case if direct current had been used to energize the cable.

Present connection at end of cable between internal conductor and armor is apparently satisfactory, but I believe that the resistance of the return path through earth, water and armor, may be greatly reduced by using special type grounding box, a tentative design of which is shown in Fig. 13. This box will offer more surface to water and earth, thereby reducing resistance at that part of the circuit, and also will protect the exposed internal conductor from corrosion.

THE NEW YORK DEMONSTRATIONS.

When the greater part of the preliminary experimental work was completed at New York, arrangements were made whereby the destroyer *Semmes* was assigned for duty in connection with the public demonstration of this system. The object of this demonstration was to bring to the attention of the commercial world the possibilities of this system as an important aid to navigation, especially for the navigation of harbors during foggy or stormy weather.

On October 6th to 9th, inclusive, 1920, demonstrations were given to representatives of various radio companies, shipping interests, pilots' associations, governmental bureaus, naval

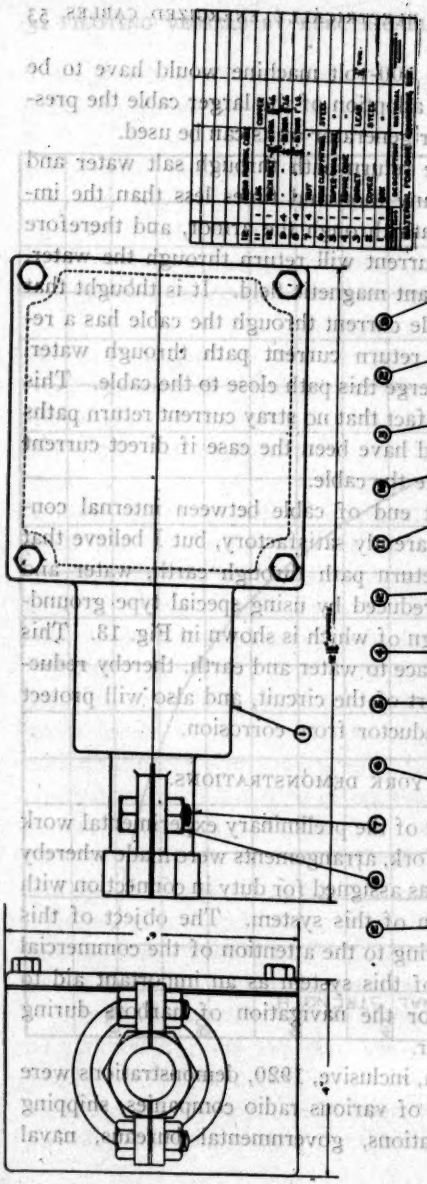


FIG. 13.

attachés and others interested in this system. During these demonstrations the various representatives were given an opportunity to inspect the apparatus and personally pilot the vessel by means of this system.

On each day of the demonstration, during the return trip to the city, a representative of either the Pilots' Association or steamship company, volunteered to pilot the *Semmes* through the Ambrose Channel. In order that the person volunteering for this duty could better demonstrate the workings of this system, the bridge was blanked with strips of canvas so that it was impossible for the person to note which direction the ship was being conned. Four persons volunteered for this work, all being able to follow the cable, and at no time were they more than one hundred yards from the cable. On Saturday, October 9th, 1920, Captain Battle, of the Cunard liner *Virgilia*, volunteered to pilot the *Semmes* back to port, and gave an excellent demonstration of the possibilities of the system. Captain Battle was a total stranger to the system, but after a little coaching, piloted the *Semmes* with such accuracy that he kept the vessel at a distance of 50 yards from the cable, and on the correct side of the channel. This was accomplished by striving to keep the *Semmes* in such a position so as to obtain a certain signal strength from the port coil, which operation was entirely practicable.

Captain Battle's work shows the feasibility of the use of one cable, although certain persons present, technical men, thought that two cables would be better, one cable to be energized with 500-cycle current, and the other with 1,000-cycle current, one cable to be used for outgoing traffic and the other for incoming traffic. One hundred and fifty persons were present at these demonstrations, and approximately ninety per cent were in favor of the single cable. All present were convinced of the practical possibilities of the Audio Piloting Cable, and had nothing but praise for the system.

Among the persons present, interested in the system, from a technical standpoint, were Messrs. F. Lowenstein, E. F. W. Alexanderson, G. H. Clark, G. H. Lewis, H. L. Curtiss and H. Buckley. The writer had many interesting discussions with the above-mentioned persons on the technical and practical details of the system, from which discussions much valuable data was obtained.

Special effort was made by the writer to obtain the ideas of numerous seafaring men on the practical possibilities of the system. The majority of the persons interviewed said that the cable should be laid beyond the lightvessel, to a distance of at least five miles, at which place vessels could make good use of the cable, and thereby pick up the lightvessel, the sight of which would serve a psychological purpose of giving the navigator more faith in the system and also a point from which to set a new course.

The proposed method to be followed in using the cable is to pick up the cable at sea and follow it to the lightvessel, from which point, by means of a stop watch, speed chart and the cable, it will be possible when making standard speeds to know when the ship is approaching the turns in the channel, and thus be doubly assured of safely navigating the ship into the anchorage near the Narrows.

FUTURE WORK.

The problem presented to date has been apparently met with the aural receiving equipment, but there is a demand for some kind of a visual device which will indicate directly the location of the cable. Work is now being undertaken by the New York Navy Yard on the design of such a device.

There are many improvements to be made in the design and installation of the coils, amplifiers, condensers and switching devices, which comprise the aural receiving equipment. Work on this equipment is being done by the New York Navy Yard in order that Naval vessels can be supplied with this equipment.

It is proposed to build a power unit similar to Fig. 14, which will eliminate the use of the Omnigraph transmitter and the relay. This unit will consist of a 4-H.P., 220-volt, 2 or 3 phase induction motor, a 2-KW. 250-volt, 500-cycle generator, an exciter for supplying direct current for generator fields, and a commutator for sending characteristic signals into the cable.

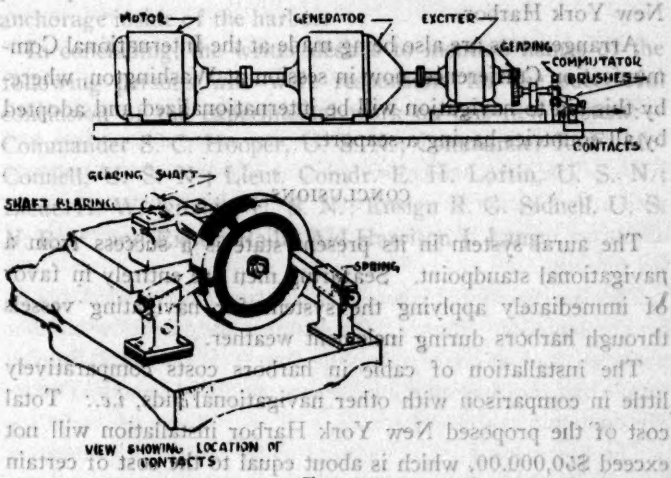


FIG. 14.

The generator or exciter field rheostats can be used in place of a series regulating device in the cable circuit, by use of which rheostat the current flow into the cable can be efficiently varied within desired limits.

The cable for the New York Harbor will most likely be laid to a point at least one mile beyond the lightvessel, from which point it will divide and be laid in northeasterly and southerly directions, for a distance of approximately five miles. Such a combination of cables could take care of transatlantic and northern bound traffic. The northeasterly cable could be an additional safeguard to keep vessels from running aground on Long Island.

The Bureau of Lighthouses, Department of Commerce, is now co-operating with the Navy Department with a view towards taking over the installation and maintenance of future cables. At present the Superintendent of the Third Lighthouse District is working with the Commandant of the Navy Yard, New York, in order to become familiar with the system and also to assist in the installation of the permanent cable for the New York Harbor.

Arrangements are also being made at the International Communication Conference, now in session at Washington, whereby this aid to navigation will be internationalized and adopted by all countries having a seaport.

CONCLUSIONS.

The aural system in its present state is a success from a navigational standpoint. Seafaring men are entirely in favor of immediately applying the system for navigating vessels through harbors during inclement weather.

The installation of cable in harbors costs comparatively little in comparison with other navigational aids, *i.e.:* Total cost of the proposed New York Harbor installation will not exceed $50,000.00, which is about equal to the cost of certain individual beacons and buoys now used in and about the New York Harbor.

When vessels of the Cunard, White Star and other steamship lines are detained outside of the harbors, due to inclement weather, the hourly maintenance cost per vessel is from $500.00 to $4,000.00. Figuring the cost of receiving equipment for the Audio Piloting Cable system at approximately $1,000.00 per vessel, it can be seen that the equipment will pay for itself within a period of a few hours. It is also thought that the insurance companies will give lower rates to vessels fitted out with this equipment, which will be an additional benefit to the ship owner.

The advantages of the system are numerous and open the way to greater safety at sea, especially when this system is used in connection with the present chain of U. S. Naval Radio Compass Stations. Vessels when approaching the coast can obtain radio compass bearings and from these bearings steer a course for the mouth of the harbor, at which place they will pick up the cable and follow a course over the cable to the anchorage inside of the harbor.

In concluding, the writer desires to mention names of the following persons who were responsible for the successful conclusion of the experimental work on this new system:—Commander S. C. Hooper, U. S. N.; Commander R. F. McConnell, U. S. N.; Lieut. Comdr. E. H. Loftin, U. S. N.; Lieut. H. W. Kitchin, U. S. N.; Ensign R. G. Sidnell, U. S. N. R. F., and Expert Radio Aid Harrison J. Lang.

FUEL OIL FOR THE NAVY.

EXTRACT FROM ADDRESS BY HON. JOSEPHUS DANIELS, SECRETARY OF THE NAVY, AT THE BANQUET A. S. N. E., DECEMBER 18, 1920.

The country which has an assured supply of oil is the nation which is best equipped for an efficient merchant marine and Navy. The time has passed when a coal burner can compete with an oil burner in commerce. No coal burner can fight an oil burner on anything like equal terms. Oil is the very life of sea fighting.

The late Admiral Fisher, of the British Navy, declared in his autobiography, that— "It's criminal folly to allow another pound of coal on board a fighting ship." He told the English people that they had an immense sure supply in Canada, Persia, Mesopotamia and elsewhere.

The first use of oil in fighting ships was in a gunboat in commission in the war between the States. On the 7th of March, 1913, just before the beginning of the larger naval program, the Navy asked the Interior Department for information as to the future supply of oil because of its importance in the policy of building oil-burning ships. The answer was that the Navy could "rely upon the reserves already existing for a supply of fuel oil for a period greater than the life of any battleship to be constructed within the next decade." The great pre-war and war construction have been built exclusively for burning oil. Of course, the great war drain upon America was not foreseen. "The war was won on a sea of oil," and most of the oil for the allied fleets came from the United States and Mexico. Since the war the price has soared, because of

world demand for American oil, so high that at times in order to secure American oil for American naval ships it has been necessary to commandeer it.

Other nations are conserving their oil in the ground and buying American oil. We have been prodigal to the point of injury to American interests on the sea. We should conserve our own oil here and obtain reserves in other countries. We should have oil storage wherever American naval or merchant ships touch. This country should not put an absolute embargo upon oil, but there is crying need for such restrictions as will not jeopardize the future needs of the Merchant Marine and the Navy. In an emergency an adequate supply of oil for our fighting ships might turn the tide of battle. We should in these quiet days see to it that this supply is conserved.

President Taft and President Wilson set aside oil reserves for the Navy. For seven years it has been necessary for the Navy Department to be vigilant to prevent the exploitation of these reserves by trespassers who had no legal right to the oil set aside for naval use. The long contest ended by legislation at the last session of Congress which more than recognized every equity. I have been compelled to fight almost every day of my incumbency in office to prevent the dummy entrymen and illegal operators from taking the naval reserve, the only hope for the Navy when the all too rapid use of American oil will leave it the only available supply. Naval needs are paramount. Without oil our Navy would be like painted ships upon a painted ocean. The use of oil in fighting ships increases naval efficiency one-third. Its utilization has revolutionized naval warfare almost as much as the construction of the *Monitor*, the big gun ship, and smokeless powder. If our Navy burned coal and another Navy burned oil the advantages would be all on the side of the oil burner.

The United States produces sixty-one per cent of the world's supply of oil, but we are consuming more than we produce, and our imports from Mexico this year will be 140,000,000

barrels. It is estimated that all the oil will be exhausted in seventeen years. This makes it all the more important for the Navy to conserve its oil reserves. Not a week passes but finds pressure to lease oil reserves and the demand for turning these reserves over to present operations increases. The Navy Department in answer to all appeals has the answer of John Paul Jones: "I have just begun to fight." In the future this demand will increase, and the arguments will be plausible, but it has cost too much grim determination to hold on to them for the Navy to permit any plan to let this last resource of the Navy to be diverted. As long as I am Secretary of the Navy, I will never consent to the loss of any portion of the Naval Oil Reserve, and as long as I live in public or private life will continue to serve my country by opposing the well organized determination to obtain these reserves.

What is the present duty of our country with reference to oil? It is, first of all, to recognize that, coal and water power should be more and more utilized on industrial plants ashore and on railroads, and the oil held for ocean commerce and the Navy. If the future oil supply is as small as many believe, why should not the Government control the distribution of oil so as to insure that its merchant ships and naval ships, practically all oil burners, shall never be found impotent because of the only fuel which can insure their superiority?

We have been wasteful of our resources. Coal and oil and water power constitute the basis of national prosperity and international commerce and the best naval operation. Has not the time come when oil and coal and water power shall be nationalized, and their ownership and distribution be regulated for the good of all and not permitted to be controlled by those who have not been free from profiteering? I believe that the ownership of God-created essentials should belong to all the people, that they should be used for national needs, that private owners should be justly compensated, but that steps

should be taken by which wasteful production, large sale abroad to the injury of America, and profiteering upon private and industrial consumers shall be ended.

The statesmanship of our day should redeem the folly of the earlier days when these prime necessities were left unregulated to our lasting injury. Whether the Government shall buy and carry on production of coal, oil and water power, or regulate their production, distribution and price is a question of pressing importance. Certainly Government cannot sit still and permit continued profiteering and waste! It must make a way or find one to guarantee to industry, our merchant marine and the Navy an adequate and continued supply of these prime necessities. Government regulation of the price of oil, coal and water power should become as fixed a government policy as regulation of railroads, express and telegraph rates. It is indeed more important as to coal and oil because they are of limited supply and the world is threatened with the peril of their exhaustion.

DEVELOPMENT OF ENGINEERING TALENT IN THE NAVY.

ADDRESS OF MR. W. L. R. EMMET AT THE BANQUET, A. S. N. E., DECEMBER 18, 1920.

In approaching the subject upon which I have been asked to speak to this society, I fully appreciate that naval conditions are likely to be misjudged by civilians, and that the conditions of life in private employment are very different from those in the service of the Government. While I have been in the Navy and have had much to do with it, my knowledge is of civil life and of methods which are in many respects not applicable to the Navy.

My associations have led me to think much about the similarities and differences of civil and military employments and to consider the extent to which the successful methods of one might be advantageously applied to the other.

In such governmental work as that of the Navy, expenditures, employments, and methods of procedure, are necessarily governed by law, authority is definite and orders must in most cases be obeyed, there is necessity for rotation in office which may cause the personnel involved in the largest undertakings to be frequently shifted. A man to be a good naval officer must be a very good all-round man, and if he devotes himself to some particular undertaking for a long time he may disqualify himself for general advancement in the service. There are, of course, many exceptions to the limitations here suggested, but that they exist seems apparent to those who are experienced in civil employments.

In civil life very great authority is exercised by certain individuals sometimes for very long periods of years, but influence and authority are constantly shifting, governed by the force of the individual and his general success. The mainspring of human efficiency lies in the freedom of one individual to trust another. And in civil life authority is based more upon such trust than upon any definite assignment of rank. Many men who fill positions of much power and influence are by no means good all-round men, although they may be invaluable in their particular line and in co-operation with the group of associates who know their abilities and their limitations.

The basis of Engineering is knowledge, and in our changing times, the basis of success in engineering is new knowledge. While accuracy and skill are important the whole history of engineering shows that the progressive spirit is more important and that great success lies, not in doing more work, but in finding better methods. Humanity has powers of lavish production and when new and more direct paths are opened they will be followed by multitudes.

If we would organize successful engineering activities we must provide scope for the innovators, and fields in which they can evolve the faculties of research and invention, because such faculties are developed by use, like others forms of art and knowledge.

The Navy of today is peculiarly dependent upon engineering and much of its engineering requirement is radically different from that applied to other activities. The Navy should not be satisfied to simply use engineering knowledge which is developed elsewhere. Many of the Navy's needs create a commercial demand which calls forth the best efforts of private concerns, while other equally important needs may be of no possible interest to outsiders. In fact, it may be a matter of very direct interest of those outsiders who are equipped to do the Navy's work to avoid novelties which may introduce uncertainty without affording prospect of increased profit.

There are two distinct fields of activity in engineering—namely, that of the Designing Engineers, and that of the Operating Engineers. The function of the latter is to keep the wheels turning in the most economical manner possible with the facilities available. To attain the best standards of success he must have knowledge, pluck and endurance of the highest type. Every naval officer should be an engineer of this kind and it has been a matter of much pride and satisfaction to me to observe what a very large proportion of our naval officers meet this specification.

The requirements of a designing and constructing engineer are different; he must have experience of the possibilities and dangers or must acquire it for the particular purpose which he has in view. His function is to go just as deep as the occasion requires, whether to do so requires one year or ten. His failures all involve financial loss and his scope of activity is generally dependent upon his success and the absence of expensive failures.

In many fields of activity the designing engineer cannot work effectively alone, but to accomplish the best results must be surrounded by tried and experienced associates and by the men and facilities necessary to elaborate experimentation. The best calculations and mathematical studies relating to engineering can only be made expeditiously by men who are constantly doing such work, and furthermore, such specialists should be men of character and imagination, working in an atmosphere where their talents are highly valued, not mere drudges working for small salaries without personal appreciation.

Creative work is seldom done by men specifically assigned to the task, but is generally self-imposed in possible conflict with other activities and in the face of more or less opposition; thus to produce such work it is desirable to create the atmosphere and prepare the field. This is what has been deliberately done in such an institution as the General Electric Research Laboratory.

A consideration of these conditions leads me to believe that the Navy should be provided with three facilities which it does not now fully possess.

1. An establishment designed for scientific and mechanical research and sufficiently equipped so that it could operate efficiently upon naval problems with such assistance as could be obtained from Navy Yards and private concerns.

2. A limited number of high grade men who desire to make engineering and science their profession and who might be available for service in all departments of the Navy where engineering is done.

3. A fund of considerable amount in proportion to construction expenditures of the Navy. This fund to be available at the discretion of the Secretary of the Navy for research and development purposes only.

My idea is that regular officers should still be used for most of the engineering work of the Navy in the research department as well as in others. The best engineers are those who have had much operating experience and a combination of practical officers with high grade engineering experts would be a happy one.

In recruiting such a corps of engineers as I have suggested it would be better to take men from the Navy or elsewhere who have had operating experience and who have shown aptitude for engineering.

Of every hundred boys who are educated for engineering, only two or three become real engineers, although many go into activities more or less associated with engineering. For this reason it is not expedient to recruit an engineering corps exclusively from men whose only known qualification is that of scholastic attainment.

One of the main advantages foreseen in the creation of such a permanent engineering group is that certain of its members would obtain through their work degrees of influence and authority in which they would be generally considered as safe

guides in the beginning of large undertakings and in the approval of expenditures. They and other officers engaged in engineering should be encouraged to connect themselves with engineering societies and to become affiliated as much as possible with engineers and scientists in civil life. Their pay should be reasonably good and the conditions of their employment should be agreeable and dignified, they should not be debarred from sea service where it might be professionally advantageous to them, and I think that they might be made eligible to hold high ranks in the Navy in cases of particular fitness, although my idea is that these men should not seek duty which is purely executive.

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JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS.

FEBRUARY, 1913—MAY, 1920.

By Commander N. H. Wright, U. S. N., Member.

Note.—The attached topical index includes the principal articles of interest to marine engineers published or reprinted in the Journal of the Society of Naval Engineers from February, 1913, to May, 1920.

It was prepared for personal use as a reading reference in connection with General Specifications for Machinery, U. S. Navy, and the main body therefore follows the grouping of General Specifications. Articles which could not be properly listed thereunder are given under supplemental headings. Following each item is given the author and a brief statement of the nature of the article.

GENERAL GROUPING.

A – V	General specification groupings.
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NOTES.

THE PROBLEM OF THE HULL AND ITS SCREW PROPELLER.*

By Rear-Admiral C. W. Dyson, U. S. Navy.

1. *Introduction.*—In 1915 Sir Archibald Denny made the statement that "in the future the rules for the correct designing of propellers should be derived from data carefully taken from the trials of smooth bottom vessels carefully run over accurately measured deep water courses."

Several years ago Admiral Taylor wrote to the effect that if theoretical formulas for design are adhered to, it would be necessary to compare each formula with experimental results and select that one which seemed to agree more closely. Then, using this as a semiempirical formula, with constants deduced from experiments or experience, problems could be satisfactorily dealt with.

Both of these statements practically agree. Denny boldly casts all theoretical treatment of the propeller into the discard, while Taylor, by implication, takes the same action, only retaining enough of the theory to save its face, to use a Chinese expression.

To accept these statements of such eminent authorities at their full value throws us at once into the field of empirical results from which to derive data and formulas for the determination of hull forms and characteristics of propellers to fit any particular form of hull and the resistance which it may be desired to overcome.

2. *Difficulties Encountered in Obtaining Reliable Data.*—Upon entering the empirical field for propeller data, the investigator is at first encouraged by the large, apparently large, amount of data available, but it does not require any extravagant amount of time for him to ascertain that fully 95 per cent of the available data is absolutely worthless, this statement of course referring to conditions where general formulas and constants for design covering the entire field of the combined problem, ships and their propellers, are being sought.

The principal causes of worthless data are:—

(*a*) Lack of model-tank trial curves for the ship.

(*b*) Variations in conditions of ship's bottom.

(*c*) Deep sea trials in place of measured mile.

(*d*) Variation in displacement and trim from those of model.

(*e*) Measured mile trials run in a tideway with insufficient number of runs for each spot.

(*f*) Trials run in shallow water.

(*g*) Model hull, when tried, not fitted with external appendages.

All of the above relate to the hull, and the responsibility for them rests with the owner, the builder and the naval architect, all of whom, apparently, are so short-sighted as to be able to see only the few dollars saved today by comparatively petty economies, losing sight entirely of the thousands of dollars which would be realized in the future were the small trial savings of today ignored.

*Read at the twenty-eighth general meeting of the Society of Naval Architects and Marine Engineers, held in New York, November 11 and 12, 1920, with some additional notes by the author.

Turning now to the propellers we find:—

(*h*) Blade forms varying according to the taste of the designing engineer.

(*i*) Blade sections varying to the same degree as blade forms.

(*j*) Variations of pitch in a blade and in the blades of the same propeller.

(*k*) Blades abnormally thick and blunt.

(*l*) Blades exceedingly rough.

(*m*) Blades made with pitch expanding radially or axially, or both.

(*n*) Hub of such form as to cause abnormal eddy losses.

(*o*) Propellers so located with respect to the hull of the ship as to produce abnormal losses.

The guilt for all of these rests with the designer of the propeller. The naval architect is, however, *particeps criminis* in the last-named offense.

3. *Derivation of Fineness of After Body of the Ship.*—In making the selection of data upon which to base formulas and design data, it is necessary to confine consideration to hulls of fine lines, with propellers so placed that the chances of decrease in propulsive efficiency due to hull effect are reduced to a minimum. The propellers which are fitted to these hulls must all have practically the same radial distribution of the projected areas, must have the same general forms of blade sections, must be true to pitch, fine-edged and smooth, in order to eliminate as many variables as possible from the equations.

VARIATIONS IN HULLS OF VESSELS.

Suppose a vessel to have a certain set of forward and of after-body lines with a parallel middle body of a certain percentage of length of the vessel. She will have a certain block coefficient. Now, maintain the same forward and after-body lines but modify the length of the middle body. As the length of the middle body increases over the initial length, the block coefficient increases, and vice versa; but the after-body lines, which control the character and quality of the flow of water to the propeller, have remained as at first. It is impossible, therefore, to base propeller conditions on block coefficient, but there must be provided a means of bringing measurements of block coefficient to a standard condition.

In the foregoing example the constants, in addition to the forward and after lines, are the beam, B, and the draught, H, while the ratio $B \div L_s$ L_s W. L., or $B \div L_s$ B. P., is variable.

Having obtained a means of correcting for the variable, there will immediately be discovered another condition which contains the same variable as before but, in addition, H has become variable. Experience has taught that with constant form and area of load-water plane, to decrease the draught of the hull is to increase the nominal block coefficient, but the wake will not increase but will remain practically the same as with the initial draught; while should the draught be increased the nominal block coefficient will be decreased but, as in the opposite case, the wake will remain practically unchanged.

Having corrected for varying ratios of length to beam and of draught to beam, it might be considered that the coast had been cleared and that a position had been reached where the propeller could be taken up, but this is not the case. There is still a variable, and that a very important one, to take care of, namely, varying ratios of lengths of after body, or if preferred to so call it, length of run, to draught. This variable has a very strong effect on the wake, varying both the character and the quality of the flow of water to the propeller.

All of the varying dimensions now having been covered, it will be found that still another correction must be made, and that is the correction for type of hull. The types of hulls are sharply defined by the character of the after-body lines as the hull fines towards the stern, whether this fining is produced by a rapid dead rise of the bottom, with a very slow decrease in beam at the load-water line, or whether it is produced by a rapid decrease in beam. The latter type can be subdivided into two types, the first having a full and the second a fine midship section, the first tending to concave body section lines and the second to convex. In the first subtype the propellers are to a great extent, or entirely, covered by the submerged hull, while in the second subtype the propeller may be entirely outside the limits of the load-water plane. The first subtype will be spoken of here as "Hulls of Type 1" and those of the second subtype as "Hulls of Type 2."

Where the hull fines rapidly by means of a rapid rise of the bottom towards the stern the vessel is denoted as having a "Hull of Type 2."

As examples of the different types the following may be taken:

Type 1.—U. S. dreadnaughts and destroyers.

Type 2.—Vessels of the ordinary fantail form of stern such as U. S. colliers and the ordinary type of merchant ship, also submarines.

Type 3.—Vessels having the co-called cruiser stern; gunboats *Annapolis* and *Sacramento*, battleships of the *Virginia* class, battle cruisers, scouts.

In reply to the questions put to the author, the questions will be taken up in order, as follows:

First: Is it to be understood that, in determining the Slip B. C. of single screw ships of all three types, no correction for variation of midship section coefficient is to be made, and does this also apply to twin screw ships of Type 1 and 2?

Answer: No correction is made for Type 3 nor for either of Types 1 and 2.

Second: Is there no correction to be made for variation of midship section coefficient in arriving at the K block coefficient?

Answer: No correction is made.

Third: Would it not be as accurate and less complicated to use the prismatic in place of the block coefficient?

Answer: Complication would not be any the less should the prismatic in place of the block coefficient be used, as in either case we would have to apply the correction for variation of ratios of Draft to Beam, for Ratio of Length of After Body to Draft and for Length of Middle Body.

Fourth: To what type would high speed yachts, torpedo boats, and the earlier destroyers belong? The fining of the after body of these vessels is produced neither by a rapid dead rise of the after sections nor by a rapid decrease of beam, but by the raising of the keel. As the propellers of these vessels are, to a great extent, covered by the submerged hull, the inference might be that they belong to Type 1.

Answer: Practically all of these boats are of Type 1.

Fifth: What is meant by a full midship section? Is it one that is as full or fuller than the standard as shown on Fig. 1? And is a fine midship section one that is considerably finer than this standard?

Answer: A full midship section is one that is as full or fuller than the standard as shown on Figure 1. A fine midship section is one that is considerably finer than this standard. The curve of midship sections on Figure 1 fits vessels of Type 1 and Type 2.

Sixth: How would Figure 5 be applied to ships of Type 2? Although one of this type is shown in Problem No. 1, the method of getting the slip is not explained.

Answer: The curve marked "Slips for S. B. C. on X-W, Types 1 and 3 Hulls" has marked on it values of S. B. C. To find the basic apparent slip for any S. B. C. with any given ratio of length of after body to H, pass a direction curve through the point for this S. B. C. as marked on the first mentioned curve. Where this direction curve intersects the ordinate through the given ratio will be the basic apparent slip to use.

Remark: It would appear that when a propeller is to be designed for a ship, where the effective horsepower is not given but has to be estimated, or if the value of 2 L. A. B. ÷ H is not known, the original method (of design) is about as good (as the new) at least for ships of this class. For small single screw steam yachts, with a large ratio of beam to length and with small midship section coefficient, the method as shown in this paper, it is believed, would be decidedly more accurate.

Reply: Experience up to date has indicated that the second method gives approximately the same results as to power as were obtained by the original method but it is considered more accurate in determination of revolutions and in designing the screw so as to plot more surely in the safety zone.

Where the value of twice the Length of After Body divided by H is not known and we have reason to believe that the after body is not abnormally long as compared to the draft nor abnormally short as compared to the draft, the two cross curves of Fig. 3 marked for vessels of Type 2 and vessels of Types 1 and 3 can be used.

Following is given a table of ship characteristics with the type to which each vessel belongs, and as performances of different vessels become available their screws should be analyzed in order to determine accurately the type the hulls belong to and add to the table. By doing so it will be found that it will aid very much in the future, as there are many cases where it is quite difficult to determine whether a vessel is of Type 1 or Type 3, or whether she is Type 2 or Type 1, although the latter distinction is easier made than the former, as the type seems to depend very considerably on the ratio of Draft to Beam.

Generally speaking the majority of vessels whose ratio of Draft to Beam is equal to or greater than the standard given on Figure 1 for Type 1 and 2 vessels are of Type 2, those from a little below the standard given for Type 3 up to the standard given for Type 1 and 2 are of Type 1, while Type 3 range from the standard given for Type 3 and below it.

In order to determine accurately to which type any particular vessel may belong, Fig. 1B has been prepared. This figure is an isometrical chart, the three dimensions being values of H ÷ B, Nominal Block Coefficient, and coefficient of midship section. Vessels of Type 3 will all plot inside the small included prism ABCE—A′ B′C′ E′. Vessels which are in the range between Type 1 and Type 3, will all plot in the small prism ABD—A′ B′ D′. Vessels of Type 2 will plot in the prism whose limits are marked on the figure as "Type 2 Limit." Vessels plotting outside of these three defined prisms are of Type 1.

American Society of Naval Engineers

Vessel	L.W.L. or B.P.	B	H Mean	H^1	Mean B.C.	H ÷ B	2 L.A.B. ÷ H or H^1	Coef E	Type
A	450'	60'.083	26'	26'	.6266	.4327	17.51	.956	1
B	510'.2	85'.21	27'.02	27'.02	.600	.317	18.88	.978	1
C	450'	80'.22	25'.02	25'.02	.633	.312	17.99	.943	1 and 3
D	435'	75'.21	23'.75	23'.75	.656	.311	14.65	.906	2
E	528'	62'.25	27'.5	27'.5	.726	.442	13.38	.984	2
F	510'	30'.7	9'.146	9'.146	.476	.298	53.89	.756	2
G	550'	55'	15'.5	13'.5	.608	.245	40.74	.942	2
H	860'	105'.75	31'	31'	.646	.2932	27.42	.957	2
I	205'.75	19'	15'.1	15'.1	.481	.847	13.5	.846	2
J	268'.57	22'.16	14'.16	11'.08	.459	.684	24.24	.751	2
K	175'	34'.125	12'.64	12'.64	.631	.366	14.61	.875	2
L	180'	35'.35	15'.23	15'.23	.672	.574	15.61	.949	2
M	210'	40'.866	11'.5	11'.5	.507	.261	18.26	.950	2
N	168'	35'	11'.46	11'.46	.482	.518	14.66	.820	2
O	300'.5	30'.5	9'.583	9'.583	.428	.314	31.55	.767	2
P	439'.17	60'.2	28'	28'	.795	.465	11.96	.984	2
Q	245'	23'.52	6'.687	6'.687	.41	.284	36.64	.77	2
R	440'	85'	28'.625	28'.625	.797	.5112	9.065	.984	2
S	440'	66'	28'	28'	.79	.51	9.5	.984	2
T	252'	50'	12'.583	12'.583	.719	.2516	17.90	.955	2
U	250'.17	41'.67	14'.852	14'.852	.441	.356	16.83	.734	1
V	160'	25'	8'.25	8'.25	.886	.33	7.27	.980	1
W	160'	16'.063	5'.5	5'.5	.570	.331	27.55	.720	1
X	147'	16'.375	4'.5	4'.5	.447	.2626	33.95	.724	1
Y	99'.5	12'.5	3'.81	3'.41	.337	.305	24.67	.800	1
Z	200'	25'.5	7'.25	7'.25	.47	.2843	26.76	.80	2
Q	50'	9'.5	2'.5	2'.5	.5206	.263	20	.725	2
S	385'	55'	24'.5	16'.81	.7842	.4623	14.75	.967	2
C	210'	45'.1	15'	15'	.576	.288	16.15	Less than standard.	3 and 1

For Spectacle frame ships H^1 = Height above base line where tangent line to lower side of Spectacle fin would cut after perpendicular. H^1, in all cases equals vertical height of rise of after body angle directing flow of water to propeller.

For Spectacle frame ships K should be taken from Curve O_3, Fig. 5, Sheet 19.

Vessels plotting in the small prism ABD—A′ B′ D′ partake of the peculiarities of both Types 1 and 3, and to the maximum degree when plotting on the plane passing through the vertex B—B′ and the middle point of the base lines AD—A′ D′. Vessels plotting in this prism should have their S. B. C's, K's and S's determined by treating them first as of Type 1 and then as Type 3. Should a vessel plot on the bisecting plane passing through B—B′, calling K_1 and K_3 the power augments obtained by treating the vessel as a Type 1 and a Type 3, the actual K will be as follows:

$$K = \frac{K_1 + K_3}{2} \text{ and}$$

$$S = \frac{S_1 + S_3}{2}$$

Should the vessel plot midway between the mid plane of the prism and the plane AB—A′ B′, or should it plot midway between the mid-plane and BD—B′ D′:

$$K = \frac{5K_3 + 3K_1}{8} \text{ and } \frac{5K_1 + 3K_3}{8} \text{ and}$$

$$S = \frac{5S_3 + 3S_1}{8} \text{ and } \frac{5S_1 + 3S_3}{8}$$

METHOD OF MAKING HULL CORRECTIONS (FIGS. 1 AND 2).

The determination of after-body form, or, as it may be expressed, the determination of the slip block coefficent of a ship, together with the effect of the variations of ratios of length, beam and draught, can be expressed graphically. With the exception of the correction for ratio of length of after-body to draught, which correction is used rather for the correction of apparent slip rather than of slip block coefficient, the preliminary estimate of this coefficient, together with its correction for ratio of draught to beam and of length-on-load-water-line to beam, for all the various hull conditions which may be considered as of normal form, are shown on Figs. 1 and 1*a*.

On Fig. 1, the line X passes through the points whose abscissas are nominal block coefficients, while the ordinates are the ratios B ÷ L. L. W. L. The value L. L. W. L. may be replaced by L. B. P. (length between perpendiculars), without any sensible error, as the entire method must be considered as one of approximation.

For the vessels of types 1 and 2, there is given a curve of coefficient of midship section at the top of Fig. 1, for the block coefficients as given by the intersection of the diagonal through the plotted point of nominal block coefficient and the abscissa value 1 with line X. Where the coefficient of midship section for any block coefficient referred to line X, is less than the standard for this referred block coefficient as given in Fig. 1, the after-body form approaches the form for type 3.

Correction for B ÷ L. L. W. L.

Of the standard hulls chosen on which to base the chart, those of type 3 had short middle bodies, which, when eliminated, would have raised the ratio line of B ÷ L. L. W. L. to A — B, while those of types 1 and 2 would have raised this ratio line to C — D, the nominal block coefficients decreasing in both cases. Now let it be supposed that the forward and after bodies remain constant but that the length of the middle bodies be

varied. As this length increases, the nominal block coefficient will increase until at length the effect of the fineness of the forward and after bodies on the value of the nominal block coefficients will become inappreciable so that the nominal block coefficient as calculated will be unity. On the other hand, the middle body may be entirely eliminated when the nominal block coefficient will be that of the combined fore and after bodies only. These changes of nominal block coefficient will take place approximately along straight lines passing through the plotted point of nominal block coefficient and the point of 1.0 on the scale of abscissas where the ordinate B ÷ L. L. W. L. is zero. The equation to these lines is—

Nominal block coefficient on

$$X = \frac{B \div L.\ L.\ W.\ L.}{.28\ (1\text{—Nom. B. C.}) + (B \div L.\ L.\ W.\ L.)}$$

Now the line X is for S. B. C's of standard twin-screw ships of all the types, and the standard B. C. for single-screw ships of type 3 would be that where the line through the plotted position of the actual ship crosses the line W, while for type 2 and type 1 it would be at the point of crossing of the line Z.

Should the vessel have four shafts, for vessels of standard ratios of H ÷ B and 2 L. A. B. ÷ H, the slip B. C. for the outboard propellers will be the intersection of the correction line with the line X, and, for the inner propellers, the intersection with the line Y. Should the ratio 2 L. A. B. ÷ H be much greater than the standard, the above corrections will be reversed. As it takes quite a large variation in S. B. C. to make a heavy variation in revolutions, this correction for position may be neglected without serious error and the point on X only be considered.

Correction for Variations in H ÷ B.

The ratios H ÷ B for type 3 ships, corresponding to the nominal B. C.'s as given by line A—B, are given by the curves on Figs. 1 and 1*a*, marked H ÷ B, type 3, standard, while those for types 1 and 2 vessels are given by the curves on the same figures marked H ÷ B type 2, standard and H ÷ B type 1. Type 1 vessels plotting on the isometric chart in front of or below the prism A B C E — A′ B′ C′ E′ must be treated as type 2 vessels in deriving the S. B. C., while vessels plotting between the plane B D—B′ D′ and Limit for type 2 on the same chart must have the S. B. C. and the Basic Slips interpolated in the same manner as for vessels plotting between types 1 and 3. There will be noted diagonal curved lines extending from above at the left to below at the right of the figures marked with the values of the nominal B. C.'s as taken from line A—B for type 3, and from C—D for types 1 and 2. These are direction lines to follow in correcting for values of H ÷ B less than standard, on Fig. 1, and for greater than standard on Fig. 1*a*.

Example illustrating the above method of estimating the S. B. C. of a vessel.

To illustrate the above, two vessels will be taken:—

Vessel A:	Nominal B. C.	=	.632
	B ÷ L. L. W. L.	=	.16
	H ÷ B	=	.3
	2 L. A. B. ÷ H	=	14.66

Examination of the after-body lines discloses that these lines fine rapidly due to a rapid falling away in beam and are convex. This indicates that the hull is of type 3 form.

The coefficient of midship section = .930.

Vessel B:	Nominal B. C.	=	.7
	B ÷ L. L. W. L.	=	.13
	H ÷ B	=	.5
	2 L. A. B. ÷ H	=	13.5
	Coefficient of midship section	=	.98

Examination of the after-body lines discloses that these lines fine rapidly due to a rapid decrease in draught of the actual full body of the hull, the beam being decreased very slowly as the stern is approached, while the screw is completely covered by the hull, in the case of twin screws. These conditions, in conjunction with any value of H÷ B indicate a hull of type 2.

Vessel A, Type 3.

Plotting vessel A on Fig. 1 with a nominal B. C. of .632 and a value of B ÷ L. L. W. L. = .16, and drawing a line through this plotted point and the point on the base line where B. C. =1.0, this line is found to cross the line A—B at B. C. = .585. Erecting an ordinate from this point on A—B to the value of H ÷ B = .3 below the curve H ÷ B, type 3, standard, and following the direction lines up to this curve, the corrected B. C. on A—B is found to be .505. Through this point .505 on A—B draw another line through the base at 1.0. Where this line crosses X, which is at an abscissa value of .585, will be the corresponding corrected value of the B. C. for wing screws, while where it crosses W, at .64, will be the S. B. C. for a single-screw vessel or for the center propeller of a multiple-shaft vessel with an odd number of shafts.

Should the vessel be fitted with wing propellers, a correction should be made for variation of midship-section coefficient from standard, by multiplying the value of the S. B. C., line X, by the ratio actual midship-section coefficient ÷ standard coefficient.

Vessel B, Type 2.

Turning now to vessel B, passing the line through its plotted point, nominal B. C. = .7, B ÷ L. L. W. L. = .13, and abscissa value 1.0, and extending the line to C—D, it will be found to cross this line at the B. C. value of .582. On the curve marked H ÷ B, condition 2, standard, the value of H ÷ B is found to be .423, therefore the actual value of H ÷ B being greater than this standard, turn to Fig. 1*a* and plot the point H ÷ B = .5, B. C. = .582, above the lower standard H—B curve, and from this plotted point follow the curves of direction down to this standard line. It will intersect this line at B. C. = .675. Through B. C. = .675 on C—D and abscissa unity, again draw a line. Where it intersects X, at B. C. = .697, will be the S. B. C. for twin propellers, while where it intersects Z, at .797, will be the S. B. C. for a single or center propeller vessel.

Vessel C, Type 1.

In vessels of type 1, the determination of S. B. C. is made as for type 2. With vessels of this type, the character of the wake appears to be nearly similar to that for type 2, while quality is more nearly like that of type 3.

By "character of wake" is meant the degree of obliquity of flow of the water to the propeller disc, while by "quality of wake" is designated the degree of disturbance of the wake and general decrease in its density due to contained air in the water in which the propeller operates.

Conditions of wake are the most favorable with ships of type 1, less so with ships of type 3, and decidedly unfavorable with ships of type 2, although it may be claimed that vessels of this latter type carry much heavier wakes than do the other types and more gain should therefore be realized from the wake than with the other two types. This claim would be a just one were the quality of the wake with the vessels of this type, with which designers ordinarily have to deal, equal to that experienced with vessels of the other types.

It has been attempted in the foregoing to give a slight idea of the difficulties in arriving at the very foundation of the screw-propeller problem before the propeller can even be approached. It is hoped that it has also shown the absurdity of basing any propeller design upon nominal block coefficient without correction. In connection with this the author had the rather amusing experience of being approached by a gentleman with the question, as to how he "would determine the slip block coefficient of a vessel to use by looking at her in the drydock." The same gentleman also made the statement that he was able to determine by inspection the proper diameter and pitch of propeller for any given hull.

It will now be necessary to say a few words concerning the resistance to be overcome by the propeller in driving the hull through the water.

RESISTANCE—HULL AND APPENDAGES.

The question of hull resistances being so fully understood, it is only necessary to state that, for propeller work in developing a system for design purposes only such vessels should be used as have had their resistances determined by towing the model of the hull in the Model Basin.

It does not suffice to tow the model of the bare hull, but this model should be fitted with all external submerged attachments in order that the true resistance of the hull may be arrived at.

Figure 2 shows curves of appendage resistance in percentage of bare hull resistance, erected on values of speed of vessel $\div \sqrt{L.}$ L. W. L., as prepared by the writer from the reports of trials of models in the model tank at Washington. These curves are rough approximations, as the actual appendage resistance varies with the form of hull and the amount of power carried by the hull and also with the revolutions at which the propellers are to run, these latter two factors fixing the size of shafting, external lengths of shafting and size of propeller struts.

PROPULSION LOSSES DUE TO TYPE OF HULL AND TO LOCATION OF PROPELLER IN RESPECT TO THE HULL.

The Power Augment Factor "K."

It is a little premature to speak of these losses at this stage of the investigation as they really cannot be studied until the primary design curves have been arrived at and this result can only be reached by selecting for the early investigations such vessels as, from the high values of the propulsive coefficients obtained, are decided upon as having all such losses reduced to the smallest possible minimum.

The losses, other than those due to poor workmanship in the propeller itself, arise from the reduction in pressure in the area in which the propeller operates, and also from the obliquity of flow of water to the propeller caused by the fullness of lines of the hull interfering with the flow. There may also be serious losses due to eddying around improperly formed stern and rudder posts, around struts of improper section and arrangement, around and between strut bosses and propeller hubs, around

the hub itself, and to excessive roughness of surface of the propeller blades and excessive variation of pitch between different blades of the propeller or throughout each blade. Of course these latter named losses are due to poor workmanship and can easily be avoided by care in casting, finishing and setting the propeller blades.

The loss factor, or "power augment," is given the letter "K" to denote it. In previous works of the author this letter "K" has been spoken of as the thrust deduction factor, but this does not convey all that "K" represents, which, in fact, are the losses which have just been given.

The K factor has the minimum values for hulls of type 1, but these values for this type increase very rapidly as the tip clearance diminishes after a certain limit of tip clearance between the hull and the circumference of the propeller disc has been reached, this distance, so far as the author can determine, being dependent upon the draught of the vessel and the height of this minimum distance above the base line. The formula for obtaining this mean tip clearance is expressed by M. T. C. $= \frac{H + 10\,H^1}{11}$ where H^1 (wing screws) = (Actual tip clearance $\times$ 10) $\div$ (Actual height of hub center above base line) and H = (actual immersion of upper blade tip in feet $\times$ 14) $\div$ (actual height of tip above base line, in feet.)

Hulls of type 3 stand next in order of loss, while those of type 2 are the heaviest losers, single-screw ships of type 3 having the same loss factor as twin-screw ships of type 2.

The values of K for the different classes of hulls are shown on Fig. 3, the ordinates being mean referred tip clearances for type 1, while the abscissa values are values of K block coefficient, K. B. C., which are obtained from Fig. 1 by taking the intersection of the line through the plotted point Nominal B. C.—B $\div$ L. L. W. L. and the abscissa point of unity, with X, W or Z, according to whether the vessel is a wing or single-screw ship. The B. C. value of this intersection is the K. B. C. for the vessel. The values given by Fig. 3 hold very closely for all vessels of what may be considered normal dimensions. For vessels having extremely great ratios of length of after body to draught, they may possibly decrease to some extent, but there is no data of sufficient accuracy to enable this to be stated definitely.

THE SCREW PROPELLER.

The vessels which are selected to be used must all have propellers having the same radial distribution of projected area but of several different projected area ratios. The vessels selected by the author were of types 1 and 3, but of different B. C.'s and ranged from a S. B. C. of .375 to .60, from a destroyer to a dreadnought, while the projected area ratios, measured outside of a circle of two-tenths the diameters of the propellers, varied from .6 to .328.

The blades were oval with the widest part of the projection at two-thirds the radius from the center, while the blade peripheries outside the widest part, measured on the arc, were well rounded.

The ships, at trial, were just out of drydock, thoroughly cleaned and painted, and were run over the measured mile in good weather. Each point of the trial curves was obtained by at least three runs at the same or approximately the same revolutions, two runs in one direction and one in the opposite. The highest point of the performance curves was obtained by five runs.

The propellers of all the ships were of manganese bronze, machined to a true pitch and to the desired blade sections, which were the same type for all the propellers and highly polished.

The trials were run by competent and experienced men, while the data were taken and computed by trained observers and engineers. There was every reason to believe that what were collected were as reliable as could ever be obtained under actual ship conditions.

Derivation of Design Curves.

Taking the derived data and from it obtaining values of pounds indicated thrust per square inch of disc area (I. $T._D$) of the propeller, and corresponding pounds propulsive thrust per square inch of projected area (P. $T._p$), a series of curves were plotted with I. $T._D$ as abscissas and P. $T._p$ as ordinates, and intermediate curves for intermediate projected area ratios were interpolated.

Curves of corresponding tip speeds in feet per minute (T. S.), and of propulsive coefficient (P. C.) were also laid down, as were also the curves of apparent slip (S).

At one point on each of the established curves of I. $T._D$—P. $T._p$ a decided change in conditions was found to occur, the curve either taking a sudden rapid rise or beginning to fall rapidly.

The values of these points in I. $T._D$, T. S., P. C., and S were taken and plotted on values of P. A. ÷ D. A. (projected area ratio), as abscissas and the equations to the curves obtained when possible, that for I $T._D$ being—

$$\text{I. T.}_D = 28.41 \left(\frac{\text{P. A.}}{\text{D. A.}}\right)^{1.7}$$

while that for T. S. is —

$$\text{T. S.} = 38148 \left(\frac{\text{P. A.}}{\text{D. A.}}\right)^{.8365+2\left(\frac{\text{P. A.}}{\text{D. A.}}\right)}$$

Equations to the curves of P. C. and of S could not be obtained.

These curves are all shown on Fig. 4, and the conditions covered by them are denoted "Basic Conditions" and are constant with the exception of the S curve which, it will be noted, is based on S. B. C. as abscissas and not on P. A. ÷ D. A. In other words, each standard S. B. C. has its constant standard basic slip.

This curve S is again shown on Fig. 5, but is there based on values of 2 × length of after body of the ship ÷ the draught, which brings in the final correction for hull form already mentioned.

Correction of S for Varying Values of 2 L. A. B. ÷ H.

On Fig. 5 is shown a curve marked "Slips for S. B. C. on X—W, Type 1 and 3 Hulls." This curve is in reality the same as the curve of S shown on Fig. 4, but in the case of Fig. 5, it is erected on abscissa values of 2 L. A. B. ÷ H instead of on S. B. C., and the S. B. C. values are marked on it at the standard values of 2 L. A. B. ÷ H as determined from Fig. 1.

Experience has demonstrated that as the lengths of after bodies of ships increase or decrease from the standard, the basic values of S decrease or increase and, in decreasing or increasing, follow a general law which may be expressed by the following equation—

$$A\,(1-S)^a = \frac{2\,\text{L. A. B.}}{H}$$

where A is 2 × the length of after body ÷ H at which the basic apparent slip becomes zero, while the exponent a depends for its value upon A. The lower scale of abscissa is the scale of A upon which the curve of the exponent, a, is erected. The curves developed from the equation for coincident values of A and a are erected on values of 2 L. A. B. ÷ H as abscissas, the scale being that given on the upper scale of abscissas, Fig. 5, while the ordinates are values of basic apparent slip, S.

The curves are direction curves according to which S varies as the ratio 2 L. A. B. ÷ H varies from standard. Thus, suppose the S. B. C. of the vessel be .575. By examination of Fig. 5, it is seen that the curve marked A = 22.5 crosses the line of standard S at S. B. C. = .575 and 2 L. A. B. ÷ H = 16.3. Now suppose the ratio 2 L. A. B. ÷ H for the vessel in question be 17½. The curve A = 22.5 is seen to cross the ordinate of 17½ = 2 L. A. B. ÷ H, at .113, which is the value of S to use in determining the basic condition of the vessel's propellers.

Computation for Basic Condition of Propeller.

To illustrate the method of determining the basic condition of a propeller, let an example be taken, as follows—

Suppose the corrected S. B. C. of the vessel be .575; the ratio 2 L. A. B. ÷ H be 17.5; the propeller blades of standard form and three-bladed.

Let Diameter = D = 16 feet
Pitch = P = 18 feet
Projected area ratio = P. A. ÷ D. A. = .34.

Then from Fig. 4 we obtain the following data:—

1. Tip speed for P. A. ÷ D. A. = .34 = 7,480 feet, from which, since Pitch ×

$$\text{Revolutions} = P \times R = \frac{\text{T. S.} \times \text{P.}}{\pi D}$$

$$\text{Basic speed} = V = \frac{P \times \text{T. S.} (1 - S)}{\pi D \times 101.33}$$

$$= \frac{18 \times 7480 \times .887}{\pi \times 16 \times 101.33} = 23.45 \text{ knots}$$

2. Indicated thrust per square inch of disc area = I. $T._D$ = 4.7 for P. A. ÷ D. A. = .34, then Basic indicated horsepower for a three-bladed propeller =

$$\text{I. H. P.} = \frac{P \times R \times \text{I. } T._D \times D^2}{291.8} = \frac{\text{T. S} \times P \times \text{I. } T._D \times D}{\pi \times 291.8}$$

$$= \frac{7480 \times 18 \times 4.7 \times 16}{\pi \times 291.8} = 11{,}045$$

and assuming that the shaft horsepower equals 92 per cent of the indicated in well-built and adjusted engines, a figure which experience demonstrates to be nearly correct, Basic shaft horsepower = S. H. P. = .92 × I. H. P. = 11,045 × .92 = 10,161.4.

3. The propulsive coefficient for a projected area ratio of .34 is seen, from Fig. 4, to be P. C. = .667, therefore the Basic effective horsepower = E. H. P. = I. H. P. × P. C. = 11045 × .667 = 7367.

Variation of the above for four blades and for two blades.

Suppose the propeller should have four blades with a total projected area ratio = .453 or a two-bladed one with a projected area ratio = .227, then—

Blades	4	2
Total P. A. ÷ D. A. =	.453	.227
P. A. ÷ D. A. =	.34	.34

that is, the basic P. A. ÷ D. A. equals ¾ total for the four blades and equals 3/2 total for the two blades.

The values of T. S. and I. $T._D$ are those pertaining to the basic P. A. ÷ D. A., while the propulsive coefficients which are dependent upon the total area ratio will be those corresponding to the total projected area ratios.

The formulas for basic I. H. P. become, for four blades:—

$$\text{I. H. P.} = \frac{\text{T. S.} \times \text{P.} \times \text{I}\,T._D \times \text{D}}{.865 \times \pi \times 291.8} = \frac{7480 \times 18 \times 4.7 \times 16}{\pi \times 252.41} = 12768.$$

for two blades:—

$$\text{I. H. P.} = \frac{\text{T. S.} \times \text{P} \times \text{I. T.}_D \times \text{D} \times .75}{\pi \times 291.8} = \frac{7480 \times 18 \times 4.7 \times 16}{\pi \times 389.1} = 8283.$$

The basic effective horsepowers now become:—

Four blades; E. H. P. = 12768 × .568 = 7252.
Two blades; E. H. P. = 8283 × .712 = 5898.

Should the blades be of other than standard distribution of projected area, other corrections occur which the limits of this paper will not allow of discussion.

Conditions other than Basic (Fig. 6, Curve 4.)

Usually the performance of a vessel and its propeller will be very remote from the basic conditions of the propeller. In such cases it is necessary to have a ready means of converting from basic to actual conditions, and such means, so far as power is concerned, are provided as follows:—

Let e. h. p. be the effective horsepower required by a given vessel for a speed v, then—

$\frac{\text{e. h. p.}}{\text{E. H. P.}}$ = fraction of basic load and $\frac{v}{\text{V}}$ = fraction of basic speed.

Taking a large number of vessels having hulls of such form and propellers so located that the value of the power augment K will be unity, and from the curves of performance of each vessel obtaining the I. H. P. required to deliver fractional loads, with the speeds v at which these fractional loads were required and denoting this I. H. P. by I. H. $P._p$ the equation for I. H. $P._p$ will be:—

$$\text{I. H. P.}_p = \text{I. H. P.} \times \left(\frac{v}{\text{V}}\right)^w$$

in which the factor $\left(\frac{v}{\text{V}}\right)^w$ is constant for any given value of $\frac{\text{e. h. p.}}{\text{E.H.P.}}$

This equation may be expressed thus:—

$$\text{Log I. H. P.}_p = \text{Log I. H. P.} + w\,(\text{Log } v - \text{Log V})$$

which, by transposing, becomes—

$$w\,(\text{Log V} - \text{Log } v) = \text{Log I. H. P.} - \text{Log I. H. P.}_p$$

Now represent this value w (Log V — Log v), by the symbol Z_p and, after establishing several values of it at corresponding load fractions, determine the equation to the curve passing through these points and the basic point where $\frac{\text{e. h. p.}}{\text{E. H. P.}}$ and $\frac{v}{V}$ both equal unity.

This equation will be—

$$Z_p = 1.0414 - 1.0414 \text{ Log } \left(\frac{10 \text{ e. h. p.}}{\text{E. H. P.}}\right)$$

or to simplify it,

$$Z_p = 1.0414 \text{ Log } \left(\frac{\text{E. H. P.}}{\text{e. h. p.}}\right)$$

for values of $\frac{\text{e. h. p.}}{\text{E. H. P.}}$ less than unity, and

$$Z_p = 1.0414 \text{ Log } \left(\frac{\text{e. h. p.}}{\text{E. H. P.}}\right)$$

for values of $\frac{\text{e. h. p.}}{\text{E. H. P.}}$ greater than unity; these values are subtractive from Log I. H. P. in the first case and additive in the second so that for values of

$$\frac{\text{e. h. p.}}{\text{E. H. P.}} < 1, \text{ Log I. H. P.}_p = \text{Log I. H. P.} - Z_p$$

$$\frac{\text{e. h. p.}}{\text{E. H. P.}} > 1, \text{ Log I. H. P.}_p = \text{Log I. H. P.} + Z_p.$$

Should the efficiency of the combined hull and propeller be such that the value of the power augment factor "K" is greater than unity, then denoting the total power by I. H. P.$_d$,

$$\text{I. H. P.}_d = K \times \text{I. H. P.}_p = \frac{K. \times \text{I. H. P.}}{10^{Z_p}}, \text{ or}$$

$$\text{Log I. H. P.}_d = \text{Log I. H. P.} + \text{Log K} \mp Z_p.$$

These equations neglect all correction for the effects of cavitation which will be discussed further on.

Problems Illustrating Analysis of Propeller.

In order to illustrate all that has been treated in the foregoing, two problems are here given, the first being the series of vessels used by Commander McEntee in his article on the "Propulsive Efficiency of Single-Screw Cargo Ships" which was read before this Society at its 1919 annual meeting.

It must be remembered, however, that in Commander McEntee's vessels the ships were not evenly balanced in their lines, the effect of shifting the middle body aft having its greatest effect on that vessel having the resultant shortest after body. As the curves on Fig. 1 for estimation of S. B. C. are based on forward and after bodies of equal length, the result may be a slight under estimate of S. B. C. and of K in the case of one or more of the vessels of the fuller nominal block coefficients.

Problem 1: Vessels of Type 2, Single Screw.

L. B. P. = 400'	$\frac{2 \text{ L. A. B.}}{H}$ =	111.9	111.4	109.88	104.37	91.86
B = 57'.3						
H = 26'	Nom. B. C.	.7242	.7438	.7634	.7830	.8026
$\frac{b}{\text{L. B. P.}}$ = .1432	K. B. C.	.758	.773	.786	.801	.815
	K	1.175	1.2	1.23	1.27	1.31
$\frac{H}{B}$ = .454	S. B. C.	.7925	.813	.825	.84	.855
	S	.14	.155	.17	.193	.2175
D = 16'.583						
P = 14'.75						
$\frac{\text{P. A.}}{\text{D. A.}}$ = .267						
Blades = 3						
T. S. = 6230						
P×R = 1764						
1 − S = .86	.845	.83	.805	.7825		
V = 14.97	14.71	14.45	14.01	13.8		
I. T.D = 3.09						
I. H. P. = 5137						
P. C. = .696						
E. H. P. = 3575						
S. H. P. = 4726						

v = 11	11	11	11	11	
$v \div V$ = .7348	.7479	.7614	.785	.8076	
e. h. p. = 1160	1200	1260	1370	1580	
e. h. p. ÷ E. H. P. = .3244	.3357	.3525	.3832	.442	
Z_p (Fig. 6) = −.51	−.492	−.47	−.432	−.382	
K = 1.175	1.2	1.23	1.27	1.31	
S. H. P.$_d$ = 1716	1827	1970	2220	2569	
Act. Power = 1750	1850	1980	2200	2700	

By comparison of the estimated and actual powers it will be seen that the only difference of serious amount is with the vessel having the heaviest nominal B. C., as was to be expected due to the excess fullness of the after body lines over those of an equally balanced ship of the same nominal B. C. This difference existing is, however, too great to be credited entirely to this cause.

There is another explanation for the excess of actual power over estimated in the case of the fifth vessel, and this is, in all probability, the correct one. It will be noted that the value of K in this case is 1.31 while that of Z_p is —.382. Taking .382—Log K as the value of Z_p corresponding to the total work, (load fraction), being delivered by the propeller, the value $\frac{\text{e. h. p.}}{\text{E.H.P.}}$ corresponding to this is found from Fig. 6, to be .555.

Plotting this point on Fig. 7, with $v \div V$ = .8076 as ordinate, it will be found to fall below the curve of S = .2175 for vessels of type 2, slips of the second order. Now assume that the existence of a K loss produces

earlier cavitation and that the propeller in the case in question is cavitating. The value of $\frac{e.\ h.\ p.}{E.H.P.}$ corresponding to the point on the curve S = .2175 whose ordinate is $\frac{v}{V} = .8076$, is approximately .525, and the value of Z_p corresponding to this is .29, then the increase log factor due to cavitation is .29 —(.382 —.11727) = .02527 and the corrected value of I. H. $P._a$, taking cavitation into account, will be:—

$$\text{Log I. H. } P._{ac} = \text{Log I. H. } P._a + .02527 = 3.43503.$$

I. H. $P._{ac}$ = 2723, which corresponds very closely with the actual power, 2700.

Now K, undoubtedly does produce earlier cavitation; therefore, when deciding upon the design point on Fig. 7, it should be so located that the $\frac{e.\ h.\ p.}{E.H.P.}$ point, corrected for K, will fall well above the basic curve of S for the vessel. The remainder of the vessels in the McEntee problem plot above their curve of S when the correction for K is made, and therefore are free from any cavitation.

Assuming that the above is correct, as it undoubtedly is, the supposition that model tank trials give no indication of cavitation is incorrect, as in the case just discussed, the model propeller is clearly indicating "Dispersal of the Thrust Column," a term which will be explained later.

Problem 2: Vessel of Type 3.

L. L. W. L. = *	Blades = 3
B = *	Form = Standard
H = *	T. S. = *
$\frac{H}{B} = .269$	P × R = 5772
	I — S = .9225
$\frac{B}{L.\ L.\ W.\ L.} = .124$	V = 52.55
	I. $T._D$ = *
$\frac{2L.\ A.\ B.}{H.} = 30.09$	I. H. P. = 225740
	P. C. = .525
Nom. B. C. = .577	E. H. P. = 118510
B. C. on A — B = .418	S. H. P. = 207680
B. C. on A — B cor. for $\frac{H}{B} = .34$	v = 32
	$\frac{v}{V} = .609$
Slip B. C. on X = .45; S = .0775	
K. B. C. = .51	e. h. p. = *(bare hull) = *(est. all append.)
K = 1	$\frac{e.\ h.\ p.}{E.\ H.P.}$ = *
Number of propellers = 4	Z_p = *
D = *	K = 1
P = *	S. H. $P._a$ = 155380
$\frac{P.\ A.}{D.\ A.}$ = *	Act. power = Between 150000 and 155000

*The data omitted are confidential and therefore cannot be made public.

Having illustrated the method followed in estimating the power required to drive a vessel at any given speed against the resistance existing at that speed, it remains to explain the method of estimating the revolutions at which the propeller will turn under these same conditions of estimated power and speed, and this will be taken care of now under the subhead—

Estimate of Revolutions Under Other than Basic Conditions of Operation.

Commander S. M. Robinson, whom the author had the pleasure of calling his assistant during the period of the Great War, by laying down curves of revolutions of propellers of standard form with actual speeds through the water as abscissas, derived a curve of exponents, y, for which the equation is as follows:—

$$y = \frac{v}{v-1}\left(2.626 + \frac{5861.3}{28045 + (v-25)^4}\right)$$

A curve of Log y was also erected on v as abscissas and the values of the logs of basic V^y and v^y were denominated Log A_V and Log A_v.

The equation for apparent slip at any speed was found to be

$$s = S\frac{\text{I. H. P.}_d \times V^y}{\text{I. H. P.} \times v^y} = S\frac{K\,V^y}{10^{Z_p} v^y}, \text{ or}$$

$$\text{Log } s = \text{Log S} + \text{Log I. H. P.}_d + \text{Log } A_V - \text{Log I. H. P.} - \text{Log } A_v$$

$$\text{Log } s = \text{Log S} - Z_p + \text{Log } A_V - \text{Log } A_v \text{ when K} = 1$$

Now let Log A_V — Log A_v = Z_S, and plotting it on values of $\frac{v}{V}$ as shown by curve 1 on Fig 6, it will be found to be expressed by the equation—

$$Z_S = 2.861 \text{ Log}\left(\frac{V}{v}\right) - .0788.$$

This curve reaches the value $Z_S = 0$ at a value of $\frac{v}{V} = .9385$ and this value of o is retained at least until $\frac{v}{V} = 1$ is reached. Beyond that the question of cavitation enters and certain variations may occur, depending upon the values of $\frac{v}{V}$ and the corresponding values of $\frac{\text{e. h. p.}}{\text{E. H. P.}}$.

The form of the equation for apparent slip, where K = 1, now becomes

$$\text{Log } s = \text{Log S} - Z_p + Z_S \text{ and when } Z_p = Z_S$$

s = S, that is, when the vessel is loaded down so that at any fraction of load $\frac{\text{e. h. p.}}{\text{E. H. P.}}$, and corresponding speed v, when Z_S for $\frac{v}{V} = Z_p$ for $\frac{\text{e. h. p.}}{\text{E. H. P.}}$, the apparent slip has become equal to the basic slip and the propellers are passing into the cavitating zone, with or without loss, depending upon the quality of the wake which the vessel carries.

To illustrate the method of estimating revolutions when cavitation does not exist, the calculations for Problems 1 and 2 will be continued.

Estimate of Revolutions.

Problem 1. $v = 11$

S. H. P.$_d$	1716	1827	1970	2220	2569
$\frac{v}{V}$	.7348	.7479	.7614	.785	.8076
Z_s for $\frac{v}{V}$ (Fig. 6—1)	+.298	+.280	+.25	+.217	+.182
s	.1010	.1142	.126	.1510	.1798
$R_d = \frac{v \times 101.33}{P \times (1-s)}$	84.07	85.31	86.46	89	92.14
Actual revolutions	84.5	85.5	86	88	93.2

Problem 2. $v = 32$

S. H. P.$_d$ 155,380 s .1965

$\frac{v}{V}$ .609 R_d 209.6

Z_s +.53 Actual revolutions 208.+

Before leaving the subjects of estimation of power and of revolutions it should be of interest to note that none of the vessels covered by the above problems was used to any extent whatever, in the derivation of the curves of design and estimating.

The processes of estimating under normal conditions having been explained, it is now necessary to take up the same subject under abnormal conditions of load, which abnormal conditions are treated under the general head of—

Cavitation (Fig. 7).

It has been shown how vessels can be divided into three general types according to the form of the after-body lines. They may also be divided into types according to the values of the basic slips, namely:—

1. Vessels having basic slips of the first order, which include vessels having basic slips of about 13 per cent and lower.

2. Vessels having basic slips of the second order, which include all vessels having basic slips exceeding about 13 per cent.

These distinctions according to the basic slips are encountered with vessels of type 2, and, in order to explain them, attention is called to Figs. 6 and 7.

Many years ago, on the trials of the British destroyer *Daring*, difficulty was experienced in obtaining the contract speed, due to failure of the propellers, such failure being accompanied by heavy vibration. The difficulty was finally overcome by increasing the projected area of the propellers to quite a large extent. After a thorough study of the phenomenon, Mr. S. W. Barnaby, of Thornycrofts, advanced the theory that the failure was due to a lack of flow of water to the propeller caused by insufficient supply head, this failure resulting in the formation of cavities in the water in which the propeller was working. He gave the name of

"Cavitation" to this action and fixed a limit of effective thrust, 11½ pounds per square inch of projected area, as that at which cavitation could be expected to occur. Later experience has demonstrated that this limit is not correct but that it depends upon the projected area ratio; the higher this ratio the higher the effective thrust before cavitation happens. Taylor supposes that both tip speed of the propeller and effective thrust enter into the solution of the problem. By inspection of Fig. 4, curves of design, it will be noted that tip speed increases with indicated thrust and with effective thrust as the projected area ratio increases. It is also a fact that, in changing from the basic conditions of resistance, for equal powers the indicated thrusts vary inversely as the tip speeds. Therefore, with a vessel loaded down very heavily beyond the designed condition the tip speeds will, as a rule, be less than the designed tip speeds and yet, as experience has taught, "Cavitation" may occur.

It is not the intention of this paper to advance any theories as to cavitation, but rather to have the student form his own theories after having had pointed out to him the various phenomena that actual performance data have shown to occur.

For convenience in discussion, the phenomena encountered where the load fraction $\frac{\text{e. h. p.}}{\text{E.H.P.}}$ is less than unity will be spoken of as "cavitation of" or "dispersal of the thrust column" of the propeller, while for those phenomena met with where the load fraction exceeds unity the term "cavitation of the suction column" of the propeller will be employed.

Dispersal of the Thrust Column.

It has already been stated that vessels can be divided into types according to the magnitude of their basic slips and that vessels of type 2 exhibit the variations due to changes in this magnitude.

Turn now to Fig. 6. Without cavitation, it has already been pointed out that to obtain the value of the apparent slip for any other than basic conditions the equation is:—

$$s = S \frac{\text{I. H. P}_d \times 10^{Z_S}}{\text{I. H. P.}}$$

or Log s = Log S + Log K + $Z_S - Z_b$, where Z_S is taken from curve 1.

With hulls of type 2, at any given value of load fraction there will be a corresponding value of speed fraction $\frac{v}{V}$, at which the quality of the wake will change materially; in the cases of vessels of type 2, having basic apparent slips of the first order, two changes will occur and there will be two corresponding values of $v \div V$. The first change with such vessels will be from a basic slip of the first order to one of the second order, while the second change will be from slips of the second order to cavitating slips. The first change will occur without loss of efficiency of propulsion when the value of K equals unity. When the value of K is greater than unity, the change of slips will be accompanied by an increase in value of K and all the phenomena of cavitation will occur. When K equals unity the second change will be accompanied by a rapid loss in efficiency of propulsion.

When the first change occurs, if it were known what the new basic S were to be, it could be inserted in the equation for apparent slip in place of the original basic S, and the values of Z_S be taken from curve 1, Fig. 6, as before. In place of making this change of basic slip, available data have given the curve marked "2" from which, using the value of the original basic S, the new values of Z_S are taken.

For this curve of Z_S, the values of Z_S vary as $3.09 \log \left(\frac{V}{v}\right)$

Now assuming that when the actual apparent slip at any gross load fraction has reached the value of the basic S, apparent slip conditions are on the verge of change,

$$\text{Log } s = \text{Log } S + Z_S - Z_p + \text{Log } K$$
$$s = S,$$

$$\therefore Z_S + \text{Log } K = Z_p.$$

Turn now to Fig. 7. The abscissas of this figure are values of load fraction $\frac{\text{e. h. p.}}{\text{E. H. P.}}$, while the ordinates are values of speed fraction $\frac{v}{V}$. The point whose abscissa value and ordinate value are both unity is the point of basic condition, while diagonals passing through the origin and crossing the ordinate $\frac{v}{V} = 1$, at any value of $\frac{\text{e. h. p.}}{\text{E. H. P.}}$ are diagonals of effective thrust ratios, the ratio being equal to the load fraction $\frac{\text{e. h. p.}}{\text{E. H. P.}}$ at which the diagonal intersects the ordinate $\frac{v}{V} = 1$, and is expressed by $\frac{\text{e. t.}}{\text{E. T.}}$, in which e. t. is the actual effective thrust while E. T. is the basic thrust of the propeller.

Lay down the curve where $Z_S = Z_p$, one of the points of which must be the point of basic condition, and it will be found that the curve, marked A, will follow the law—

I. H. P.$_p$ or S. H. P.$_p$ varies as $3.09 \text{ Log } \left(\frac{v}{V}\right)$ or Z_p varies as $3.09 \text{ Log } \left(\frac{V}{v}\right)$, which is the same as that for Z_S when $s = S$.

Calling the new basic slip of the second order, S_1, the ordinate of any point on A, $\frac{v}{V_1}$; the value of Z_S for $\frac{v}{V_1}$ taken from curve 1, Fig. 6, Z_{S_1}; the original basic slip, S, the corresponding value of the ordinate $\frac{v}{V}$; and the value of Z_S for $\frac{v}{V}$, taken from curve 2, Fig. 6, Z_{S_2} we have

$$\text{Log } S + Z_S = \text{Log } S_1 + Z_{S_1},$$

from which for any assumed values of S_1, $\frac{v}{V_1}$ with its corresponding value of Z_{S_1}, can be found a curve of values of S for a series of assumed values of $\frac{v}{V}$ (each less than $\frac{v}{V_1}$), with the corresponding values of Z_S. Taking a series of values of S_1 for any value of $\frac{v}{V_1}$, and it will be found that as S_1 increases S will increase and $\frac{v}{V}$ diminish until S_1 equals approximately 21 per cent, when the value of S will become constant for any further decrease of $\frac{v}{V}$.

The values of $\frac{v}{V}$ and the values of S at which the latter becomes constant are the limits at which the change from basic slips of the first order to those of the second order occurs, and this statement is borne out by all the data available. The curve of these slips is shown at the lower part of Fig. 7, while the limiting curve of $\frac{v}{V}$ is shown, marked B. For type 1 and type 3 vessels, this shift of Z_S values does not occur, so far as can be ascertained, but Z_S is taken always from curve 1, Fig. 6.

These two types and type 2 of first order of apparent slips having K = 1 pass on down to $\frac{v}{V}$ values shown by curves C, Fig. 7, before genuine "Dispersal of the Thrust Column," with consequent loss in efficiency, is encountered. These curves, C, have been determined from the performances of a four-shaft vessel when steaming with only one propeller in use on each side of the ship, the idle propellers dragging, a slight correction having been allowed in the e. h. p. curve of the vessel for the increase in resistance due to these idle screws. The actual increase occurring is one that should be solved by the Model Tank.

Turning now to the hulls of type 2 with basic apparent slips of the second order, only one change occurs in the slip conditions, and that is the change from slips of the second order to those of cavitation. These latter are augmented (similar to those of types 1, 3, and of 2 when original slips are of the first order, when $\frac{v}{V}$ is less than those of curves C, Fig. 7), on entering cavitation due to the increase in power demanded by the cavitating condition.

As the curve B is the change limit for slips given by the slip curve D, it is assumed that a corresponding change from slips of the second order to those of cavitation will occur at values of $\frac{v}{V}$ varying inversely as the ratios $\frac{1 - S \text{ of B curve}}{1 + S \text{ of 2d order}}$, and by assuming different values of these slips of the second order, curves of cavitation limits varying from S = .27 down to curve B have been laid down.

(When the basic slip for any type of vessel exceeds 26 per cent, the value of Z_S for any value of $\frac{v}{V}$ must be taken from curve 2, Figure 6).

(When a vessel's propellers enter into cavitation of the thrust column a radical change in conditions occurs; these are expressed by the following formulas:

Let I. H. P.$_c$ = total power required, including cavitation loss,
and I. H. P. = Basic power of the propeller, then
Z_{pgc} = Log I. H. P. — Log I. H. P.$_c$

$\frac{\text{e. h. p.}_c}{\text{I. H. P.}_c}$ = load fraction corresponding to Z_{pgc}

$\frac{v_c}{V}$ = value of $\frac{v}{V}$ where ordinate $\frac{\text{e. h. p.}_c}{\text{I. H. P.}_c}$, Fig. 7, cuts cavitating curve of S for type vessel under consideration.

$v_e = v \times \frac{v_e}{V}$

s_e = apparent slip at point $\frac{v_e}{V}$ and $\frac{\text{e. h. p.}_e}{\text{E. H. P.}}$

$s_e = S \times \frac{\text{I. H. P.}_d \times 10^{Z_{Se}}}{\text{I. H. P.}}$

Z_{Se} = Speed factor corresponding to $\frac{v_e}{V}$

s = actual apparent slip to be expected at the speed v,

then $s = \frac{v_e - v}{v_e}(.95 - s_e) + s_e$

$R_d = \frac{101.33 \times v}{P \times (1 - s)}$

This correction has not been applied in this article but should have been).

For hulls of type 2, curve B has been checked by the performances of submarines of the Holland type, while the curves for slips of the second order have been checked by submarines of the Lake type. These two types usually are radically different in the forms of hulls and in the arrangement of their propellers. Vessels of the Holland type require the use of the surface draught in obtaining the S. B. C., while only one-half diameter of the hull is used in determining the ratio 2 L. A. B. ÷ H, and those of the Lake boats, while obtaining the S. B. C. in the same manner as for the former, use a much greater value than one-half diameter of hull in obtaining the after-body length ratio; in fact, for both types the H in the ratio 2 L. A. B. ÷ H equals the height of the point of the stern above the base line.

This results in a much greater value for the after-body ratio of the Hollands than for the Lakes, with a corresponding reduction in the basic slip values. Due to the position of the propellers in relation to the hull and also to the peculiar form of hull, the Lake boats have a shifting value of K depending upon the length of the vessels until this length equals or exceeds 230 feet, when K becomes unity for all surface conditions of speed. For all lengths of vessels of the Holland type operating on the surface, K has a constant value of unity.

When vessels of either type are running submerged it becomes necessary to bring the diving rudders into operation in order to maintain proper balance, and the additional resistances introduced by the resistances of the diving rudders appear to vary with the speeds, disappearing entirely at 10 knots with boats of the Lake type but reappearing at speeds above 10 knots. With vessels of the Holland type, the resistance due to the influence of the diving rudders appears to reach a maximum at 7 knots, remaining practically constant in its effect on K until 9 knots is reached, when it begins to fall rapidly.

In both types of vessels the value K at any speed appears to depend upon the length of boat; in the case of the Holland boats, K in the submerged condition becomes unity when a load-water-line length of 200 feet has been reached, while with the Lake type the value of unity is not reached until a load-water length of 230 feet has been attained.

These values of K for both types of vessels are given in the following tables:—

VALUES OF K FOR SUBMARINES.

Lake—Submerged.

L.W.L.	6	7	8	9	10	11	12
	K.						
150	1.35	1.24	1.14	1.05	1.0	1.1	1.4
160	1.30	1.20	1.12	1.04	1.0	1.09	1.34
170	1.25	1.17	1.10	1.04	1.0	1.07	1.28
180	1.20	1.13	1.08	1.03	1.0	1.06	1.22
190	1.14	1.10	1.06	1.02	1.0	1.04	1.16
200	1.09	1.06	1.04	1.015	1.0	1.025	1.11
210	1.05	1.035	1.02	1.01	1.0	1.02	1.05
220	1.00	1.00	1.00	1.00	1.0	1.0	1.0
230	1	1	1	1	1	1	1
250	All below = unity up to 12 knots.						
260							
270							
280							
290							
300							

Lake—Surface.

L.W.L.	6	7	8	9	10	11	12	13	14	15	16	17
	K											
150	1 17	1 13	1 085	1 04	1 00	1 027	1 095	1 14	1 19	1 23	1 29	1 45
160	1 145	1 12	1 075	1 038	1 00	1 023	1 083	1 112	1 17	1 21	1 25	1 35
170	1 125	1 097	1 064	1 033	1 00	1 02	1 073	1 105	1 143	1 175	1.215	1.3
180	1 105	1 082	1 055	1.03	1 00	1 019	1 062	1 089	1 12	1 147	1.18	1 25
190	1 086	1 068	1 045	1 025	1 00	1 017	1 05	1 072	1 097	1 12	1 148	1 2
200	1 067	1 052	1 036	1 02	1 00	1 013	1 04	1 055	1 075	1 09	1 11	1.13
210	1 047	1 037	1.025	1 015	1.00	1 01	1.03	1 04	1 052	1.063	1 075	1.1
220	1 025	1 020	1 013	1 01	1 00	1 01	1 015	1 02	1 03	1.035	1.04	1 05
230	1	1	1	1	1	1	1	1	1	1	1	1
250	All below = unity until $\sqrt[3]{L} = 1.12$, when there may possibly occur a rapid increase in *K*, reaching 1.22 at $\sqrt[3]{L} = 1.19$ and rising rapidly.											
270												
280												
290												
300												

Holland Type.

Submerged only as $K = 1$ for Surface.

v	5	6	7	8	9	10	10.5	11
L.W.L.	*K*							
150	1.275	1.385	1.395	1.395	1.395	1.375	1.34	1.25
160	1.12	1.23	1.25	1.25	1.25	1.21	1.19	1.07
170	1.04	1.12	1.14	1.14	1.14	1.1	1.09	1.0
180	1.0	1.04	1.06	1.06	1.06	1.03	1.025	1.0
190	1.0	1.005	1.01	1.00	1.00	1.00	1.00	1.00
200	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00

All greater lengths of L. W. L. have $K = 1$.

These *K*'s are not true power augments but are genuine variable changes in resistance, and therefore are not treated as the *K* in Problem 1, in the determination of cavitation.

To illustrate the changes occurring with type 2 vessels with true K = 1, in passing, first from slips of first order to slips of the second order, and second from slips of the second order to cavitation, analyses of the performances of two vessels are given. The first vessel used is a Holland submarine having a basic slip of the first order, the problem being divided into two parts:—(*a*) Surface condition; (*b*) submerged condition:

VESSEL C.

Hull.		*Propeller.*	
L.L.W.L.	= 268.57′	D	7′
B	= 22.16′	P	6.25′
H	= 14.16 (surface)	Blades	3
$H_1 = \frac{B}{2}$	= 11.08′	Projec. form	Standard
Nom. B.C.	= .459	$\frac{P.A.}{D.A.}$	.4
$\frac{B}{L.L.W.L.}$	= .0825	T.S.	.8520
$\frac{H}{B}$	= .684	1 − S	.875
L.A.B.	$= \frac{L.L.W.L.}{2}$	V	20.91
$\frac{2L.A.B.}{H_1}$	= 24.24	I.T.D	6
S.B.C.	= .451	I.H.P	4879 (two propellers)
S	= .125	P.C.	.62
K	= 1	E.H.P	3025
		S.H.P	4489

Surface Performance.

$\frac{e.h.p.}{E.H.P.}$	e. h. p.	v	$\frac{v}{\sqrt{L}}$	Z_0	K	S.H.P.$_d$	Act. S.H.P.$_d$	Z_0, curve 1, Fig. 6	S	e	Revs. Est.	Revs. Actual
.05	152	8.44	.404	−1.355	1	198		+1.043	.125	.0609	143.7	145
.075	228	9.57	.458	−1.1713	1	302		+.89	.125	.0654	166	166
.10	304	10.5	.502	−1.0414	1	408		+.77	.125	.0669	182.5	182.5
.15	456	11.86	.567	−.858	1	623		+.62	.125	.0723	207.3	207.5
.20	608	13.05	.624	−.7279	1	840		+.497	.125	.0735	228.4	228
.30	912	14.87	.711	−.5445	1	1281		+.335	.125	.0772	261.3	258.5
.40	1216	16.14	.772	−.4144	1	1729		+.233	.125	.0823	285.2	281.5
.50	1520	17.3	.827	−.3135	1	2181		+.154	.125	.0866	307.1	303.2
.60	1824	18.5	.885	−.231	1	2637		+.070	.125	.0863	328.3	328
.70	2128	19.36	.926	−.1613	1	3096		+.019	.125	.0901	345	347
.80	2432	20	.957	−.1009	1	3558		+.0	.125	.0991	359.9	363
.90	2736	20.5	.980	−.0477	1	4022		+.0	.125	.112	374.3	375

Submerged Condition.

e. h. p. / E.H.P.	e. h. p.	p	$\frac{v}{V}$	Z_p	K	S.H.P.$_d$	Act. S.H.P.$_d$	Z_p curve 2, Fig. 6	S	s	Revs. Est.	Revs. Actual
.025	76	5.05	.242	−1.667	1	96.7	105	+1.64	.125	.1862	100.6	102
.05	152	6.44	.308	−1.355	1	196.2	199	+1.575	.125	.2075	131 8	129
.075	228	7.33	.351	−1.1715	1	302.5	280	+1.4	.125	.2116	150.7	146
.10	304	8.06	.386	−1.0414	1	408	364	+1.275	.125	.2141	166.3	161
.15	456	9.36	.443	− .858	1	622.5	550	+1.09	.125	.2138	190.8	186
.20	608	10.26	.491	− .7279	1	840	750	+ .95	.125	.2085	210.2	208
.30	912	11.85	.567	− .5445	1	1281	1170	+ .76	.125	.2053	241.8	235.5
.40	1216	13.19	.629	− .4144	1	1729	1620	+ .622	.125	.2016	267.1	262

Unfortunately, in the case of this vessel no measure of shaft horsepower on the surface was taken, the vessel being driven by Diesel engines. In the submerged condition electric motors were the propelling units. Attention is called to the extremely close agreement between estimated and actual revolutions, the slips not, as yet, having changed their character from those of the first to those of the second order. In the submerged condition the actual powers were obtained by measurements of electric input and voltage to the motors and the resulting K. W. transformed to S. H. P. by dividing by the factor .746. The estimated S. H. P.$_d$ is seen to be somewhat in excess of the actual, and this excess estimate is accompanied by an excess in estimated revolutions over the actual. Should the actual measured power at the highest submerged speed be used in the calculation of revolutions for that speed, the resulting estimated revolutions become 264.1 instead of 267.1.

The vessel just discussed has an after-body of such form that it is almost at the dividing line between vessels of type 2 and type 1, and the character and quality of the wake are so good that there is no augment of power, that is, $K = 1$.

Should the vessel be one of type 2, slips of the first order, but with the character and quality of the wake such that K is in excess of unity, the performance becomes radically different. Upon entering cavitation, the value of K becomes increased to approximately $K_c = 1.26\ K_n$.

Suppose in the case of such a vessel a series of performances be taken in which the net load fraction $\frac{\text{e. h. p.}}{\text{E. H. P.}}$ is constant but the speed fraction $\frac{v}{V}$ for this load fraction is decreasing due to progressive loading of the vessel, then

$$\text{Log I. H. P.}_d = \text{Log I. H. P.} + \text{Log K} - Z_p$$

Now let

$$Z_{pg} = Z_p - \text{Log K}$$

that is Z_{pg} is the gross load factor for the propeller when delivering the net load fraction $\frac{\text{e. h. p.}}{\text{E. H. P.}}$, with a power augment K. Call the gross load fraction $\frac{\text{e. h. p.}_g}{\text{E. H. P}}$.

When $\frac{v}{V}$ has decreased until it reaches the intersection of the ordinate at $\frac{\text{e. h. p.}_s}{\text{E. H.P.}}$ with the cavitating curve of the basic slip S, of the first order, cavitation ensues and K changes value to $K_c = 1.26\ K$.

The equation for power then becomes

$$\text{Log I. H. P.}_d = \text{Log I. H. P.} + \text{Log } K_c - Z_p.$$

As the value of $\frac{v}{V}$ still decreases, call the value of the load factor corresponding to the point on the cavitation curve of S whose ordinate is $\frac{v}{V}$, Z_1, the equation for power becomes

$$\text{Log I. H. P.}_d = \text{Log I. H. P.} + \text{Log } K_c - Z_p + Z_1 - Z_{pg}$$

but $Z_p - \text{Log } K_c = Z_{pg}$, therefore

$$\text{Log I. H. P.}_d = \text{Log I. H. P.} - 2Z_{pg} + Z_1.$$

As in Problem A, the slip before cavitation equals $s = S \dfrac{\text{I. H. P.}_d}{\text{I. H. P.}} 10^{Z_S}$, where Z_S is taken from Curve 1, Fig. 6, but as soon as the cavitation curve of S is passed, while the equation remains the same, the value of Z_S must be taken from Curve 2, Fig. 6.

Turning now to the second vessel, which is of the Lake type, the range of apparent slips of the second order is entered. In the estimates of revolutions both in and out of cavitation, the values of Z_S are taken from curve 1, Fig. 6, but in the estimates of power in the cavitation range a corrective factor must be introduced. Let this factor be called K_1. Its values are found as follows:

Turning to Fig. 7, attention is called to two points marked Z_p, one plotted on $\frac{\text{e. h. p.}}{\text{E. H. P.}} = .575$ and $\frac{v}{V} = .78$, while the other is on $\frac{\text{e. h. p.}}{\text{E.H.P.}} = .5$ and $\frac{v}{V} = .51$. Now for either of these points to find the value of K_1, take the value of $\frac{\text{e. h. p.}}{\text{E.H.P.}}$, called $\frac{\text{e. h. p.}_1}{\text{E. H. P.}}$, where this same value of $\frac{v}{V}$ cuts the cavitating thrust curve for the basic slip S of the vessel. The Z_p value for this value of $\frac{\text{e. h. p.}_1}{\text{E. H. P.}}$ call Z_1, then $K_1 = Z_1 - Z_p$ and

$$\text{Log I. H. P.}_d = \text{Log I. H. P.} + \text{Log K} - 2Z_p + Z_1$$

$$\text{Log S. H. P.}_d = \text{Log S. H. P.} + \text{Log K} - 2Z_p + Z_1$$

and, by transposing the equation,

$$Z_1 = \text{Log S. H. P.}_d + 2Z_p - \text{Log S. H. P.} - \text{Log K}$$

is obtained from which, when the actual power and load fraction are known, the cavitation loss can be closely approximated.

Also, by transposing, the equation—

$$2Z_p = \text{Log S. H. P.} + \text{Log K} + Z_1 - \text{Log S. H. P.}_d,$$

from which the load fraction $\frac{\text{e. h. p.}}{\text{E.H.P.}}$ corresponding to Z_p can be ascertained.

VESSEL D.

Hull.

L. L. W. L. = 203.75′
B = 19′
H = 16.1
H_1 = 15.1

$\frac{B}{L.\ L.\ W.\ L.}$ = .093

$\frac{H}{B}$ = .847

L. A. B. = ½ L. L. W. L.
2 L. A. B. ÷ H_1 = 13.5
Nom. B. C. = .481
B. C. on C—D = .35
B. C. on C — D corrected for $\frac{H}{B}$ = .52
S. B. C. on X = .55
S = .225

Propeller.

Baldes 3
Shafts 2
Form ..... Standard
D ..... 6.17′
P ..... 5.542′
$\frac{P.A.}{D.A.}$ 4
T. S. 8,530
1 — S ..... .79
V 18.65
$I.T._D$ 6
I. H. P. 3,830
P. C. ..... .62
E. H. P. 2,371
S. H. P. 3,523

Surface Condition—Analysis.

v	$\frac{e.h.p.}{E.H.P.}$	Z_p	K	S. H. P.$_d$ Est.	S. H. P.$_d$ Act.	$\frac{v}{\sqrt{p}}$	Z_s	S	s	Revs. Est.	Revs. Act.
6	.0253	1.667	1.06	80.4	80	.3217	1.32	.225	.1073	122.9	.....
8	.03075	1.32	1.042	176	200	.4289	.975	.225	.1061	163.6	160.1
10	.1131	.98	1.0	369	370	.5361	.688	.225	.1149	206.6	203.4
12	.1873	.757	1.037	639	632	.6433	.46	.225	.1177	248.7	247
14	.3122	.522	1.068	1131	1134	.7506	.2784	.225	.1371	296.7	300.5
16	.443	.362	1.10	1684	1670	.8578	.11	.225	.1386	339.6	345
17	.574	.25	1.13	2239	2500?	.9114	.035	.225	.155	367.9	362.5

The power data for the 17-knot point are doubtful as there was no way of measuring it, and the shop brake-test power for the revolutions was used.

Submerged Condition.

v	$e.h.p._D$	$\frac{e.h.p.}{E.H.P.}$	Z_p	$\frac{v}{\sqrt{p}}$	$\frac{e.h.p._1}{E.H.P.}$	Z_1	K	S. H. P.$_d$ Est.	S. H. P.$_d$ Act.	Z_s	S	s	Revs. Est.	Revs. Act.
5	.100	.04219	1.41	.2681	.05	.....	1.1	151	.....	1.55	.225	.3422	139	.....
6	.140	.05512	1.31	.3217	.067	.....	1.08	186	.....	1.32	.225	.2482	145.9	.....
8	.264	.1114	.983	.4289	.089	1.08	1.032	473	464	.975	.225	.2852	206.6	203.4
10	.476	.20	.7279	.5361	.145	.87	1.00	914	896	.688	.225	.2846	255.6	253
12	.774	.3265	.51	.6433	.255	.62	1.10	1543	1628?	.46	.225	.2842	306.5	303.5

Neither of vessels C and D was employed in obtaining the curves of design but, unfortunately, due to lack of reliable data, they both formed one of several vessels used in obtaining the values of K for their respective types.

Still another vessel, but this time of type 3, will be given as an illustration for that type.

VESSEL E.

Hull.		*Propeller.*	
L. L. W. L.	= 168′	Propellers	1
B	= 36′	No. of blades	4
H	= 11.46′	Form	Standard
B ÷ L. L. W. L.	= .214	D	9.3′
H ÷ B	= .318	P	11′
L. A. B.	= ½ L. L. W. L.	Total $\frac{P.A.}{D.A.}$	.338
2 L. A. B. ÷ H	= 14.66	P. A. ÷ D. A.	.2535
Nom. B. C.	= .482	T. S.	6,010
K. B. C.	= .65	1 — S	.865
B. C. on A — B	= .517	V	18.91
B. C. on A — B cor. for $\frac{H}{B}$	= .517	I. T.$_D$	2.8
S. B. C. on W	= .646	I. H. P.	2,218
S	= .135	P. C.	.659
		E. H. P.	14.62

ANALYSIS.

v	= 13.17	e. h. p.	= 696	Z_p = .35	I.H.P.$_d$ = 1228
$\frac{v}{V}$	= .6965	e. h. p. / E.H.P.	= .4763	K = 1.25	Actual I.H.P.$_d$ = 1208
Z_s	= .365				
s	= .1727				
R_d	= 146.5				
Act. R_d	= 146.7				

This vessel, also, was not used in obtaining any of the design curves.

To illustrate the method to be followed in making an estimate of the effective horsepower delivered by a propeller when the I. H. P.$_d$ or S. H. P.$_d$ and the speed of the vessel are known, take up the case of vessel F.

VESSEL F.—Hull of the dreadnaught type having a standard basic slip of approximately 13 per cent.

Propeller.	
Number	4
Blades	3
Form	Standard
D	13.42′

P	15.17′
$\frac{P.A.}{D.A.}$	.3567
$T.S.$	.7760
$1-S.$	.86
V	23.97
$I.T_D$	4.97
I.H.P	34,259 on four propellers.
P.C.	.648
E.H.P.	22,200 on four propellers.
S.H.P.	31,519 on four propellers.

Ship steaming with one idle propeller on each side :—

v	7	10	13.5	15
$\frac{v}{V}$	.292	.4172	.5632	.6258
S. H. $P._a$	1,760	4,443	9,263	11,586
Z_S (curve 1, Fig. 6)	1.44	1.0	.62	.49
s	.3995	.3661	.3182	.295
R_a	77.87	105.4	132.3	142.1
Act. R_a	75.9	101.9	132.1	143.85
S. H. P.	15,760	15,760	15,760	15,760
K	1	1	1	1
$\frac{\text{e. h. p.}_a}{\text{E. H. P.}}$ for $\frac{v}{V}$	.06	.122	.33	.545
Z_1	1.25	.94	.5	.27
Z_p	1.101	.745	.365	.202
e. h. p. ÷ E. H. P.	.105	.192	.442	.635
e. h. p.	1,166	2,131	4,910	7,050

In the above calculations—

$$2Z_p = \text{Log S. H. P.} - \text{Log S. H. P.}_a + Z_1.$$

It has been stated that with vessels of types 1 and 3 the transposition of Z_S from curve 1 to curve 2, Fig. 6 does not occur. This statement requires modification because it does happen when what may be called "induced cavitation" occurs. By this term is meant cavitation which is produced by means other than the conditions normally governing the performance of the propeller. It is frequently encountered with vessels of the destroyer type when propeller struts have been so constructed and fitted as to increase the squat of the hull and to produce areas of reduced pressure in the wake of the struts, the propellers working in these areas. In such cases the value of Z_S gradually passes over from curve 1 to curve 2, the departure from curve 1 for vessels of 310 feet L. W. L. length occurring approximately at a speed $v = 1.42\sqrt{\text{L. L. W. L.}}$ and arriving on curve 2 when $v = 1.92\sqrt{\text{L. L. W. L.}}$

To illustrate this phenomenon, the case of a destroyer is given in which the shafts were carried by two struts each, the axis of the lower arm of each forward strut being parallel to the base line of the vessel, while the axes of the lower arms of the after struts were inclined downward at the forward edge, 4½ degrees.

VESSEL C.

Hull.

L. L. W. L. = 300.5
B = 30.5′
H = 9.583′

$\frac{B}{L.\ L.\ W.\ L.} = .1015$

Nom. B. C. = .428

$\frac{H}{B} = .314$

L. A. B. = 150.25

$\frac{2L.\ A.\ B.}{H} = 31.35$

$B.\ C.\ on\ C - \frac{D}{H} = .345$

$B.\ C.\ on\ C - D\ cor.\ for\ \frac{\ }{B} = .271$

$S.\ B.\ C.\ on\ \frac{X}{S} = .315 \div .21$

$v_1 = 1.42 \sqrt{300.5} = 24.75$

$v_2 = 1.92 \sqrt{300.5} = 33.44$

$\frac{v_1}{V} = .85$

$\frac{v_2}{V} = 1.279$

Propeller.

Number 2
Blades 3
Form ..... Standard

D ..... 92.5″

P ..... 82″

$\frac{P.\ A.}{D.\ A.}$ ..... .595

T. S. 13,330

P × R 3762

1 − S ..... .79

V 29.33

I. T. D. 11,77
I. H. P. 18,021
P. C. ..... .525
E. H. P. 9,461

S. H. P. 15,680

$\frac{e.\ h.\ p.}{E.\ H.\ P.}$	e. h. p.	v	$\frac{v}{V}$	z_p	K	S. H. P. Est.	S. H. P. Act.	z_p	S	s	Revs. Est.	Revs. Act.
.025	237	10.3	.3512	−1.668	1	356	400	+1.215	.21	.074	165	160
.05	474	12	.4092	−1.355	1	732	720	+1.025	.21	.09823	197.3	200
.075	711	14.3	.487	−1.1715	1	1117	1150	+ .805	.21	.09093	233.3	230
.10	946	15.52	.5292	−1.0616	1	1507	1500	+ .70	.21	.09568	254.5	252
.20	1892	19.25	.6564	− .7229	1	3102	3100	+ .435	.21	.1070	319.7	317
.30	2838	21.3	.7332	− .5445	1	4732	4750	+ .30	.21	.1196	362.1	359
.40	3785	22.8	.7775	− .4144	1	6385	6300	+ .248	.21	.1432	394.6	394
.50	4730	24	.8184	− .3135	1	8055	8100	+ .165	.21	.1492	418.3	425
.60	5676	25.1	.8559	− .231	1	9741	9700	+ .122	.21	.1634	444.9	453
.70	6623	26.15	.8917	− .1613	1	11436	11500	+ .105	.21	.1845	475.5	476
.80	7570	27.15	.9258	− .1009	1	13143	13100	+ .095	.21	.2072	507.8	508
.90	8514	28.15	.9599	− .0477	1	14855	14900	+ .08	.21	.2277	540.5	538
1.00	9461	29.2	.9957	0.0	1	16580	16550	+ .061	.21	.2417	571	571
1.05	9935	29.94	1.021	+0.0221	1	17445	17400	+ .05	.21	.2479	590.3	591

*Z_S taken on straight line passing over from $\frac{v_1}{V}$ on curve 1, Fig. 6, to $\frac{v_2}{V}$ on curve 2; where curve of $\frac{\text{e. h. p.}}{\text{E. H. P.}} - \frac{v}{V}$ crosses ordinate of unity load at lower than $\frac{v}{V} = 1$ (Fig. 7), curve 2 becomes a horizontal line coinciding with $Z_S = 0$ Fig. 6. The change of the Z_S condition does not change so abruptly as is done in making the estimate, for inspection of the actual revolutions indicates that it begins at about 23 knots and changes gradually until at 26 knots it has taken its position on the pass-over line used in the estimate.

Cavitation when $\frac{\text{e. h. p.}}{\text{E. H. P.}}$ *Is Greater than Unity.*

When the load fraction exceeds unity, the laws governing the action of the propeller, both as to power and apparent slip increments, appear to change, but the changes occurring appear to be dependent upon the value of the speed fraction at which the performance plots on Fig. 7.

Should the performance plot on or above the curve E, Fig. 7, and at a load fraction exceeding unity, the values of Z_p follow the curve marked 5 on Fig. 6, while should it plot below the curve E, the value Z_p follows the curve 4, the equation for curve 4 being—

$$Z_p = 1.0414 \text{ Log } \left(\frac{\text{e. h. p.}}{\text{E. H. P.}}\right)$$

while that for curve 5 is—

$$Z_p = 1.51 \text{ Log } \frac{\text{e. h. p.}}{\text{E. H. P.}}$$

Should the performance plot above E, the values of Z_S follow the curve 3, which differs slightly in equation from the cavitation curve 2, Fig. 6, the new equation being—

$$Z_S = 3 \text{ Log } \left(\frac{v}{V}\right)$$

Should it plot below E, the values of Z_S for values of $\frac{v}{V}$ greater than .9385, become zero, the increase in apparent slips being apparently independent of speed but entirely dependent upon power.

In order to illustrate these phenomena, the performances of four vessels will be analyzed, the first, vessel H, having been used in the preparation of all the design curves, vessel I, a reciprocating engine destroyer plotting above E; vessel J, a destroyer plotting below E; vessel K, a dreadnaught plotting below E.

The trial data of vessels I and J are not of the best as the propellers of I were cavitating heavily, producing excessive vibration of the engines and affecting the clearness of the indicator cards, while with vessel J the starboard propeller and torsion meter readings suffered badly through "induced cavitation" of that propeller, the port propeller running smoothly.

VESSEL H. TYPE I.

Hull.		*Propeller.*	
L. L. W. L.	= *	Number of propellers.	2
B	= *	Number of blades....	3
H	= *	Form of proj. area...	Standard
B ÷ L. L. W. L.	= .167	D	18.25′
Nom. B. C.	= .6	P	19.75′
B. C. on C — D	= .572	P. A. ÷ D. A.....	.328
$\frac{H}{B}$	= .317	T. S.....	7,250
B. C. on C — D corrected for $\frac{H}{B}$	= .492	P × R.....	2,498
Slip B. C. on X	= .525	1 — S	.865
L. A. B.	= *	V	21.32
2 L. A. B. ÷ H	= 18.88	I. T. D	4.35
S	= .135	I. H. P.....	24,800
K	= 1	P. C.....	.665
		E. H. P.....	16,492

*The data omitted are confidential and therefore cannot be made public.

e. h. p. / E. H. P.	e. h. p.	v	$\frac{v}{\sqrt{V}}$	Z_p	I. H. P. Est.	I. H. P. Act.	Z_s	S	s	Revs. Est.	Revs. Act.
.0747	1232	9.5	.3896	−1.1718	1670	1600	+1.09	.135	.1118	49.1	52
.09957	1642	10.45	.4902	−1 042	2251	2200	+ .8	.135	.07742	58.12	57.5
1991	3284	13.21	.6196	− .728	4639	4800	+ .502	.135	.08023	73.69	73.6
2987	4926	15.05	.7059	− .545	7054	7125	+ .346	.135	.08538	84.43	84.5
.3983	6568	16.45	.7716	− .415	9538	9300	+ .235	.135	.08919	92.66	92.5
4978	8210	17.7	.8302	− .314	12035	11600	+ .145	.135	.09148	99.96	99 75
5974	9852	18.92	.8875	− .233	14503	14625	+ .07	.135	.09275	107	106.8
697	11494	19.76	.9269	− .1615	17098	17200	+ .017	.135	.09679	112.3	112.1
7965	13136	20.33	.9536	− .1012	19645	19600	+0	.135	.1069	116.8	116.2
.8961	14778	20.8	.9756	− .048	22205	22200	+0	.135	.1209	121.4	120.2
.9957	16420	21.24	.9963	− .001	24740	25200	+0	.135	.1347	125.9	124.8
1 045	17230	21.44	1.006	+ .0294*	26533	26600	+ .015	.135	.1396	127.9	126.5
1 095	18062	21.62	1.014	+ .036*	28213	28100	+ .023	.135	.1457	129.9	129.2

*In this case Z_p follows curve 5, Fig. 6, and Z_s follows curve 3.

The next vessel given is extremely interesting on account of the enormous range of load fraction and large range of speed fraction through which the performance extends.

VESSEL I. TYPE 2.

Hull.		*Propeller.*	
L. L. W. L.	= 245′	Number	2
B	= 23.52′	Blades	3
H	= 6.687′	Form	Standard
B ÷ L. L. W. L.	= .096	D	7.42′
Nom. B. C.	= .41	P	10.833′
B. C. on C — D	= .332	P. A. ÷ D. A.	.275
H ÷ B	= .284	T. S.	6400
B. C. on C — D corrected for $\frac{H}{B}$	= .247	P × R	2974
S. B. C. on X	= .295	1 — S	.7775
2 L. A. B.	= 245	V	22.82
2 L. A. B. ÷ H	= 36.64	I. T.D.	3.23
S	= .2225	I. H. P.	3625
		P. C.	.693
		E. H. P.	2512

s	$\frac{s}{V}$	e. h. p.	$\frac{\text{e. h. p.}}{\text{E. H. P.}}$	Z_p	I. H. P.$_d$ Est.	I. H. P.$_d$ Act.	Z_S	S	s	Revs. Est.	Revs. Act.
16	.7013	467	.1859	−.762	627	.....	+.35	.2225	.08617	163.8	.....
18	.789	787	.3133	−.522	1090	.....	+.21	.2225	.1085	188.9	184
20	.8767	1287	.5123	−.298	1825	.....	+.082	.2225	.1353	216.4	220
22	.9643	1978	.7874	−.105	2847	2750	+.0	.2225	.1747	249.4	255
24	1.052	2881	1.147	+.09	4460	4500	−.07	.2225	.233	292.6	296.5
25	1.096	3297	1.313	+.18	5487	5500	−.122	.2225	.2543	313.6	313.5
26	1.14	3738	1.488	+.26	6597	6500	−.178	.2225	.2687	332.6	324
27	1.184	4193	1.669	+.34	7931	7600	−.226	.2225	.2893	355.4	342
28	1.227	4474	1.781	+.38	8696	8750	−.278	.2225	.2814	364.5	368
28.5	1.249	4622	1.84	+.4	9106	9300	−.3	.2225	.2801	370.3	374

The performance of this vessel's propellers enters the overload zone to a greater extent than any other of which the records are available. The maximum load fraction attained is 1.84, while the corresponding speed fraction is 1.249. The ratio S. H. P. = .92 I. H. P. probably does not exactly fit this case as the main air pumps were driven by the main propelling engines and absorbed some of the power developed by those engines.

VESSEL J.

This vessel is a destroyer fitted with turbines connected directly to the propeller shafts. The revolutions are high with resultant high tip speeds, while the effective thrusts delivered at 35 knots were also very high. The blades of the starboard propeller had been injured by striking something and were run in this injured condition, with the result that the performance of this propeller was extremely erratic. The port propeller, however, ran smoothly to the end of the trials, and the power data taken from this shaft were consistent. In comparing the estimated and actual performances, the actual power of the port shaft has been doubled for the total power, while the revolutions of that shaft have been taken as the normal revolutions.

Hull.		*Propeller.*	
L. L. W. L.	= 310′	Number	2
B	= 30.7′	Blades	3
H	= 9.146′	Form	Standard
B ÷ L. L. W. L.	= .99	D	7.8′
Nom. B. C.	= .476	P	7.42′
H ÷ B	= .298	P. A. ÷ D. A.	.648
S. B. C. on X (corrected)	= .285	T. S.	15,100
2 L. A. B. ÷ H	= 33.89	P × R	4,572
S	= .222	1 — S	.778
		V	35.11
		$I. T._D$	13.65
		I. H. P.	26,025
		P. C.	.525
		E. H. P.	13,663
		S. H. P.	23,943

Analysis.

v	e. h. p.	$\frac{v}{V}$	e. h. p. / E. H. P.	Z_p	K	S. H. P.$_d$ Est.	S. H. P.$_d$ Act.	Z_S	S	s	R_S Est.	R_S Act.
12	430	.3418	.03147	−1.53	1	707	800	+1.255	.222	.1179	185.7	184
16	1050	.4558	.07685	−1.14	1	1735	1800	+ .894	.222	.1260	249.9	249.9
20	2210	.5697	.1618	− .82	1	3624	3900	+ .62	.222	.1401	317.4	310
24	4700	.6837	.3440	− .481	1	7910	8300	+ .388	.222	.1792	399.1	389
28	8600	.7976	.6294	− .205	1	14934	15000	+ .2	.222	.2195	489.6	468
32	12970	.9115	.9493	− .0221	1	22755	23000	+ .032	.222	.2271	565.1	563
34	15100	.9685	1.105	+ .0632	1	27495	27500	+ .0	.222	.2594	626.6	623
35	16200	.997	1.186	+ .08	1	28953	29400	+ .0	.222	.2669	652	655

These vessels begin squatting at about 20 knots and, when the struts are properly located so that no eddying with consequent reduction of pressure occurs, an increase in wake over the normal is generally encountered, resulting in a slowing down of actual revolutions below the estimate. The tip speed of the propeller at 35 knots was 16,050 feet with an effective thrust per square inch of projected area of 16.9 pounds, both figures being exceptionally high.

VESSEL K.

This vessel is of the dreadnaught type and has propellers of such characteristics as to plot her performance considerably below curve B, Fig. 7. Her hull is of type 1.

Hull.		*Propeller.*	
L. L. W. L.	= *	Number	4
B	= *	Blades	3
H	= *	Form	Standard
B ÷ L. L. W. L.	= .168	D	9.583′
Nom. B. C.	= .622	P	8.193′
B. C. on C — D	= .586	P. A. ÷ D. A.	.501
H ÷ B	= .305	T. S.	10,670
B. C. on C — D corrected for $\frac{H}{B}$	= .455	P × R	2,004
S. B. C. on X	= .49	1 — S	.84
2 L. A. B. ÷ H	= 19.5	V	24.13
S	= .16	I. T. D.	8.75
K. B. C.	= .612	I. H. P.	31,732
Mean T. C.	Large	P. C.	.554
K	= 1	E. H. P.	17,580
		S. H. P.	29,196

Analysis.

$\frac{e.h.p.}{E.H.P.}$	e. h. p.	v	$\frac{v}{\sqrt{L}}$	Z_p	K	S. H. P. Est.	S. H. P. Act.	Z_s	S	s	Revs. Est.	Revs. Act.
.1	1758	10.31	.4273	−1.0414	1.0	2654	2750	.98	.16	.139	148	153
.2	3516	12.96	.5371	− .7279	1.0	5463	5400	.68	.16	.143	187	189.5
.3	5274	14.81	.6138	− .5445	1.0	8333	8100	.515	.16	.149	215	218
.4	7032	16.3	.6755	− .4144	1.0	11244	11050	.400	.16	.158	240	241
.5	8790	17.44	.7228	− .3135	1.0	14183	13700	.317	.16	.161	257	259
.6	10548	18.42	.7634	− .231	1.0	17152	16300	.250	.16	.167	274	274.5
.7	12306	19.32	.8007	− .1613	1.0	20138	19150	.19	.16	.171	288	288.5
.8	14064	20.1	.8350	− .1009	1.0	23143	22000	.142	.16	.176	302	302.5
.9	15822	20.7	.8579	− .0477	1.0	26159	25300	.11	.16	.185	314	315
1.0	17580	21.13	.8757	− .00	1.0	29196	28300	.082	.16	.193	324	326
1.05	18459	21.3	.8848	+ .0221	1.0	30720	30000	.075	.16	.200	329	331.5

Concerning the efficiency of propulsion, results of trials indicate that with type 1 and probably very fine type 3 ships, where the basic slip is greater than the S values corresponding to curve B, Fig. 7, there is a gradual increase in efficiency in passing from the upper curve of S to B

*The data omitted are confidential and therefore cannot be made public.

and then a gradual decrease until the cavitating curve for the basic S is reached. However in making the estimates of performance, this probable increase should be omitted and carried as "so much possible velvet." The maximum gain is probably reached with basic slip between 21 and 22 per cent, becoming practically zero with those basic slips for which the upper S curves on Fig. 7 approximate closely to B or are below B.

WAKE GAIN AND LOSS.

Before taking up the subject of design, there is one phenomenon met with to which attention should be called, and this is the effect produced by a change of wake which may occur due to the vessel altering her lines of flotation. This variation of wake may be due to a change in trim, causing an increase in wake during a portion of her career, but the wake usually returning to normal after an upper limiting speed is attained, or it may be caused by the vessel lifting bodily in the water, decreasing its submerged fullness and tending to plane. This latter action is accompanied by a heavy decrease in wake and a corresponding falling off in efficiency in propulsion.

Vessels which slightly alter their trim at a speed depending upon the ratio of v to $\sqrt{\text{L. L. W. L.}}$ may accomplish this change in trim without an increase in the wake. Such a vessel is vessel H, while another ship of similar type may show a decided gain in efficiency during this period of increased wake due to the resulting increased thrust of the propeller produced by the increase in wake. Such a gain is spoken of as "wake gain." Such gain is shown by vessel L, following. Vessels H and L have effective horsepower curves practically identical in form. Both show a slight hump between 15 and 17 knots. The performances on trial of the two ships were radically different. The writer was aboard both of these vessels and noted that vessel H, which was the lighter and smaller of the two ships, after passing 15 knots developed a very peculiar vertical vibration which gave one the impression that she was trying to break her back. This vibration disappeared after passing 17 knots. Vessel L, on the contrary, passed through the range of the hump smoothly and with a total lack of this peculiarity of vibration. Analysis of the performance of vessel L shows that when the estimate of power and revolutions are made for the speeds and effective horsepowers obtained from the model tank, the revolutions obtained agree with the revolutions corresponding to the actual speeds corresponding to these estimated powers.

When a vessel lifts bodily in the water and begins planing as she lifts, the model tank curves indicate a rapid decrease in the increment of increase of the resistance. Should the estimate of power required be made for the e. h. p. and speed given on this curve of deflection, a bitter disappointment will be encountered. Rather, the curve before inflection should be extended by means of a batten and the v's corresponding to this extended curve of e. h. p. will be the actual speeds obtained by using the model tank curve of e. h. p. and v. After planing is accomplished, the wake gradually increases to normal as the speed is increased.

Gain by possible wake increase should never be counted on as a certainty, but loss by wake decrease should always be allowed for.

VESSEL L.

In the case of this vessel, the estimated powers and revolutions are those obtained from the model tank curve while there are also given columns of the actual S. H. P.s, speeds with corresponding revolutions, obtained by fairing the hump out of the model tank curve and using the new speeds as those to be expected rather than those used in the estimate.

Condition I.

Hull.

L. L. W. L.	= *
B	= *
H	= *
B ÷ L. L. W. L.	= .162
Nom. B. C.	= .646
B. C. on C — D	= .595
H ÷ B	= .207
B. C. on C—D corrected for $\frac{H}{B}$	= .455
S. B. C. on X	= .49
L. A. B.	= *
2 L. A. B. ÷ H	= 20.82
S	= .135
M. T. C.	= Large
K	= 1

Propeller.

Number	4
Blades	3
Form	Standard
D	12.863′
P	11.209′
P. A. ÷ D. A.	.4247
T. S.	9,000
P × R	2,502
1 — S	.865
V	21.61
I. T.$_D$	6.62
I. H. P.	37,404
P. C.	.603
E. H. P.	22,556
S. H. P.	34,412

Analysis.

$\frac{L}{\sqrt[3]{\Delta}}$	E. H. P.	$\frac{P}{\text{tank}}$	$\frac{v}{\sqrt{L}}$	z_p	K	S.H.P.$_d$	n_p faired	S.H.P.$_d$ for v_1	z_s	S	η	R_d Est.	R_d Act. for v_1
.1	2256	11	.5091	−1.0414	1	3128	11.23	3250	+ .75	.135	.069	106.8	102
.2	4512	13.86	.6415	− .7279	1	6439	14.0	6550	+ .46	.135	.0729	135	129.6
.3	6768	15.75	.729	− .5445	1	9822	16 12	9850	+ .305	135	.0778	154.4	151.3
.4	9024	17.25	.7984	− .4144	1	13253	17.67	13200	+ .198	.135	.082	169.9	168
.5	11280	18.4	.8517	− .3135	1	16719	18.8	16750	+ .115	.135	.0861	182	181.7
.6	13536	19.3	.8933	− .231	1	20217	19.7	20300	+ .061	.135	.0913	192	190.6
.7	15788	20.05	.928	− .1613	1	23736	20.34	23700	+ .015	.135	.0964	200.6	198.6
.8	18048	20.7	.9581	− .1009	1	27278	20.85	27250	0	.135	107	209.6	206
.9	20304	21.15	.9789	− .0477	1	30839	21.25	31100	0	.135	1131	215.6	213.5
1.0	22556	21.45	.9928	0	1	34412	21.55	Off curve	0	.135	.135	224.2	221

DESIGN OF THE PROPELLER.

In treating this subject an attempt will be made to be as brief as possible, confining the discussion entirely to the precautions that must be taken in deciding upon the initial point of the design and to the formulas used in obtaining the design.

Determination of the Design Point.

The data supplied the designer in order to determine upon the proper propeller to meet any given set of conditions usually come in one of three forms:—

1. Required speed, effective horsepower for this speed, and revolutions of propeller desired at this speed.

*The data omitted are confidential and therefore cannot be made public.

2. Effective horsepower curve for the ship, designed power of the engines and revolutions of the propeller at this power.

3. Designed power of the engines and revolutions of the propeller, desired speed of vessel.

With each of these are supplied the hull characteristics of the vessel, together with distance between center of propeller hub and skin of ship or between center of hub and limiting dimensions of the propeller well.

When data of form 1 are given, the problem consists in determining not only the characteristics of the propeller but also the engine power necessary to drive the vessel at the required speed.

When data of form 2 are given, in addition to the characteristics of the propeller, the maximum speed of vessel which can be obtained with the designed power of the engines is also determined.

Finally, with form 3 data, the propeller designer is in the hands of the naval architect. So far as the designer of the propeller is concerned, the ship may or may not make the desired speed with the designed power at the designed condition of load. He can only feel confident that she will make it at some load condition. To call on him for a guaranteed performance is, to say the least, an absurdity, and to blame him for failure is to carry injustice to the extreme.

In selecting the point upon which to base the design, Fig. 7 is used, and the preliminary point or points taken will depend upon the type of vessel and the derived basic slip of the vessel. The point or points taken should be at sufficient height above the cavitation curve for the basic slip and the value of K as to insure that for any probable increase in resistance of the hull, due to deep loading, foul bottom, or ordinary heavy weather conditions, the value of $v \div V$ will not be reduced to such an extent as to lower the performance point below the curve of cavitation.

Ratio of Pitch to Diameter.

The limitation for this ratio is not definitely known, but for safety it is recommended that a lower value of about .75 be taken as a limit, although conditions may arise where it will be found necessary to go even lower than this value. The curve E, Fig. 7, need not be adhered to rigidly as the upper limit of design, but this limit and the maximum limit of $\frac{\text{e. h. p.}}{\text{E.H.P.}}$ used for the adopted point of design should be regulated by the values of $P \div D$ and of P. A. $\div$ D. A. which are obtained in the calculation. It is undesirable to use a basic value of P. A. $\div$ D. A. of less than approximately .23 on account of the bluntness of blade sections which result with the smaller values.

Formulas for Design.

Form 1: Data as per 1.

(1) v (required speed)	v	v	v	(Const.)
(2) $n \times$ e.h.p. (req. eff. horsepower)	$n \times$ e.h.p.	$n \times$ e.h.p.	$n \times$ e.h.p.	(Const.)
(3) n (number of propellers)	n	n	n	(Const.)
(4) e.h.p. (on 1 propeller)	e.h.p.	e.h.p.	e.h.p	(Const.)
(5) e.h.p. $\div$ E.H.P. (load fractions used)	Val. 1	Val. 2	Val. 3	(Variable)
(6) $v \div V$ (speed frac. for values of (5))	1	2	3	(Variable)
(7) E.H.P. $=$ (4) $\div$ (5)	1	2	3	(Variable)
(8) $V =$ (1) $\div$ (6)	1	2	3	(Variable)
(9) S (basic slip from hull characteristics)	S	S	S	(Constant)

(10) D (proposed diameter).....	D	D	D (Constant)
(11) E. $T._p \times$ (P. A. $\div$ D. A.).....	1	2	3 (Variable)
$E.T._p \times (P.A. \div D.A.) =$	(2.88 $\times$ E.H.P.) $\div$ ($D^2 \times V$) for 3-bladed propellers.		
	(2.491 $\times$ E.H.P.) $\div$ ($D^2 \times V$) for 4-bladed propellers.		
	(3.84 $\times$ E.H.P.) $\div$ ($D^2 \times V$) for 2-bladed propellers.		
(12) P.A. $\div$ D.A. (for (11) Fig. 8)	1	2	3 (Variable)
(12) = Projected area ratio if the propeller is 3-bladed = Basic.			
(13) 4/3 (P. A. $\div$ D. A.) = (Total proj. area ratio for 4 blades)	1	2	3 (Variable)
(14) 2/3 (P. A. $\div$ D. A.) = (Total proj. area ratio for 2 blades)	1	2	3 (Variable)
(15) P. C. (propul. coef. for (12), (13) or (14).....	1	2	3 (Variable)
(16) I.H.P. = E.H.P. $\div$ P. C. = Basic I.H.P.	1	2	3 (Variable)
(17) S. H. P. = .92 $\times$ I. H. P. = Basic shaft horsepower	1	2	3 (Variable)
(18) Z_p = Power factor for $\frac{\text{e. h. p.}}{\text{E.H.P.}}$ (Fig. 6).....	1	2	3 (Variable)
(19) K = Power augment (Fig. 3).	K	K	K (Constant)
(20) $I.H.P._d$ or $S.H.P._d = (I.H.P. \div 10^{Z_p}$ or S. H. P. $\div 10^{Z_p}) \times K =$ Est. of power.....	1	2	3 (Variable)
(21) T. S. = Tip speed for (12)...	1	2	3 (Variable)
(22) 1—S = 1—(9)	(1—S)	(1—S)	(1—S) (Constant)
(23) P = (πD $\times$ V $\times$ 101.33) $\div$ (T.S. $\times$ (1—S)) = Pitch	1	2	3 (Variable)
(24) Z_S = Speed factor for $\frac{v}{V}$, Fig. 6, Curve 1.....	1	2	3 (Variable)
(25) s = (S $\times$ I. H. $P._d \times 10^{Z_S}$) $\div$ I. H. P. = (S $\times$ S.H.$P._d \times 10^{Z_S}$) $\div$ S. H. P. = Ap. slip.....	1	2	3 (Variable)
(26) R_d = ($v \times$ 101.33) $\div$ (P $\times$ (1—s)) = Est. revs.....	1	2	3 (Variable)

Should the revolutions obtained be less than those desired and a good ratio of P $\div$ D exist, the revolutions may be increased in two ways, both of which produce a reduction in efficiency.

(*a*) Retain original diameter and increase values of $v \div V$.

(*b*) Reduce the diameter and retain the original values of $v \div V$.

By the (*a*) process, the projected area ratio will be increased and the values of P $\div$ D and of *s* will be reduced, but the safety distance above cavitation will increase.

By the (*b*) process, the values of the projected area ratios will be increased, P $\div$ D will remain constant and *s* will be increased, while the safety remains as originally.

(*c*) A third method may also be used; that is, the rise above the original point may be made by keeping the effective thrust constant. This will maintain constant projected area ratios, reduced efficiency, decrease in *s* and increase in revolutions, but the reduction in efficiency will not be so great as in (*a*).

Should the revolutions obtained be higher than those desired, follow the opposite course to (*a*), (*b*) or (*c*), and the effects will be opposite. Care must be taken to preserve sufficient safety clearance above the cavitation curve, and to do this it may be necessary to hold the $\frac{v}{V}$ constant and use a lower value of $\frac{\text{e. h. p.}}{\text{E.H.P.}}$

Form 2: Data as per (2).

(1) v (covering range of expected speed)	1	2	3	(Variable)
(2) $n\times$e.h.p. (total ef. H. P. for v's)	1	2	3	(Variable)
(3) n = number of propellers.....	n	n	n	(Constant)
(4) e.h.p. = ef. H. P. on one propeller	1	2	3	(Variable)

Proceed as in form 1. Should the revolutions for the designed power be greater or less than those desired, proceed as in (*a*), (*b*) or (*c*).

The expected speed will be that corresponding to the designed power and revolutions as determined by the calculations.

Form 3: Data as per (3).

(1) n S. H. $P._d$ (designed shaft horsepower)	n S.H.$P._d$	n S.H.$P._d$	n S.H.$P._d$	(Con.)
(2) n I. H. $P._d$ (designed I. H. P.) $= n$ S. H. $P._d \div .92$.....	n I.H.$P._d$	n I.H.$P._d$	n I.H.$P._d$	(Con.)
(3) n (number of propellers).....	n	n	n	(Con.)
(4) I. H. $P._d$ (I. H. P. on one propeller)	I.H.$P._d$	I.H.$P._d$	I.H.$P._d$	(Con.)
(5) v (expected speed).....	v	v	v	(Con.)
(6) e.h.p. ÷ E.H.P. (assumed load fractions)	1	2	3	(Variable)
(7) $v \div V$ = Speed fractions at assumed points on (6).....	1	2	3	(Variable)
(8) D (assumed diameter).....	D	D	D	(Constant)
(9) S (basic slip for hull conditions)	S	S	S	(Constant)
(10) $V = v \div (v \div V)$ = Basic speed	1	1	1	(Variable)
(11) K = Power augment.....	K	K	K	(Constant)
(12) Z_p = Power factor for $\frac{\text{e. h. p.}}{\text{E.H.P.}}$ (Fig. 6).....	1	2	3	(Variable)
(13) I. H. P. = I. H. $P._d \times 10^{Z_p} \div$ K = Basic power.....	1	2	3	(Variable)
(14) I. $T._D \div (1-S)$.....	1	2	3	(Variable)

$$\text{I. T.}_D \div (1-S) = \begin{cases} (2.88 \times \text{I. H. P.}) \div (D^2 \times V) \text{ for 3-bladed propellers.} \\ (2.491 \times \text{I. H. P.}) \div (D^2 \times V) \text{ for 4-bladed propellers.} \\ (3.84 \times \text{I. H. P.}) \div (D^2 \times V) \text{ for 2-bladed propellers.} \end{cases}$$

(15) P. A. ÷ D. A. = Proj. area ratio for (14) Fig. 8.....	1	2	3	(Variable)

(15) = Projected area ratio of the propeller if 3-bladed = Basic.

(16) 4/3 (P. A. ÷ D. A.) = Total projected area ratio for 4-bladed.

(17) 2/3 (P. A. ÷ D. A.) = Total projected area ratio for 2-bladed.

(18) P. C. = Propulsive coef. for (15), (16) or (17).....	1	2	3	(Variable)

(19) E. H. P. = I. H. P. × P. C. = Basic effective horsepower..	1	2	3	(Variable)
(20) e.h.p. = E. H. P. × (e.h.p. ÷ E. H. P.) = ef. horsepower delivered	1	2	3	(Variable)
(21) p. c. = Propulsive coef. expected = n × (20) ÷ (1) or (2).....	1	2	3	(Variable)
(22) T. S. = Tip speed for (15), Fig. 4.....	1	2	3	(Variable)
(23) 1—S = 1— (9)	(1—S)	(1—S)	(1—S)	(Constant)
(24) P = (πD × V × 101.33) ÷ (T. S. × (1—S)).....	1	2	3	(Variable)
(25) Z_S = Speed factor for $\frac{n}{V}$, Fig. 6, curve 1.....	1	2	3	(Variable)
(26) s = (S × (I. H. P.$_a$ or S. H. P.$_a$) × 10Zs) ÷ (I. H. P. or S. H. P.).....	1	2	3	(Variable)
(27) R_a = (v × 101.33) ÷ (P × (1—s)).....	1	2	3	(Variable)

If proper revolutions are not obtained, proceed as before.

Attention should be called to one danger that is incurred where I. H. P.$_a$ or S. H. P.$_a$ are used in obtaining the characteristics of a propeller. This danger is produced by an incorrect estimate of K. Should the K used be too small, the propeller obtained will be too large while the efficiency promised will be greater than will be realized. The propeller will, however, be lifted higher in the safety zone than expected. Should K be too large the opposite results occur.

Propellers of Reduced Diameter.

In many problems that arise, the permissible diameter of propeller will be too small to reduce the revolutions under standard conditions of design to so low a number as the problem requires. In such cases it becomes necessary to resort to what may be called a method of "reduced diameter."

There are two of these methods, as follows:—

1. Method by changing form of blades to "fantail."
2. Method by changing form of blades to "broad tip."

By the first method, the propeller, by using the actual effective horsepower to be delivered or a modification of the propelling I. H. P.$_a$ or S. H. P.$_a$, is designed for a larger diameter and lower number of revolutions than specified. The tips are then cut off the blades and the original projected areas retained by widening the blade tips beyond the centers of pressure by an amount of area equal to that cut off from the tips. This method produces higher pitches, considerably lower revolutions and slightly less efficiency than the standard method of design and must be used when the reduction in revolutions is considerable.

By the second method, the propeller is designed for a greater diameter than can be carried, using a higher effective or engine power than required for the ship. The tips are then cut off the blades, reducing them to the diameter allowed.

This method maintains practically constant efficiency with the basic full diameter propeller, with a higher pitch and lower revolutions than would be obtained by the regular method of design, but the revolutions are not decreased to the same extent as by the first method of reduction.

Formulas for Reduced Diameter.

Form 1: "Fantail."

(1) D = Diam. possible.....	D	D	D (Constant)
(2) D_1 = Assumed diameter > D.....	1	2	3 (Variable)
(3) $D_1 \div D$	1	2	3 (Variable)
(4) $(D_1 \div D)^{\frac{1}{2}}$	1	2	3 (Variable)
(5) e.h.p (effective horsepower on one propeller)	e.h.p.	e.h.p.	e.h.p. (Constant)

Now proceed as in preceding methods, using D_1, until I. H. $P._d{}^1$ or S. H. $P._d{}^1$ have been obtained. The actual corrected power will then be—

$$\text{I. H. P.}_d \text{ (corrected)} = \text{I. H. P.}_d{}^1 \times \left(\frac{D_1}{D}\right)^2.$$

Use D_1 in obtaining the pitch, and then proceeding to the apparent slip, use the uncorrected value of I. H. $P._d{}^1$ or S. H. $P._d{}^1$. Use this slip in obtaining the revolutions for the propeller with diameter D_1, then corrected revolutions equal—

$$R_d \text{ (corrected)} = R_d{}^1 \times \left(\frac{D_1}{D}\right).$$

When the basis of design is engine power instead of effective, the power must be reduced as follows:

$$\text{I. H. P.}_d{}^1 = \text{engine power} \times \left(\frac{D}{D_1}\right)^3.$$

Form 2: "Broad Tip."

(1) D = Diameter possible.....	D	D	D (Constant)
(2) D_1 = Assumed diameter > D...	1	2	3 (Variable)
(3) $(D_1 \div D)^2$	1	2	3 (Variable)
(4) $(D_1 \div D)^{\frac{1}{2}}$	1	2	3 (Variable)
(5) e. h. $p._1$ = e. h. p. (actual $\times$ $(D_1 \div D)^2$)	1	2	3 (Variable)

Use D_1 and e. h. $p._1$ in designing the full diameter propeller. Calling the speed of ship v, then by the regular method of design, using e. h. $p._1$ for v, the derived power for the full diameter propeller required for the speed v will be I. H. $P._{d1}$ and the actual power required for the reduced diameter propeller will be—

$$\text{I. H. P.}_d = \text{I. H. P.}_{d1} \times \left(\frac{D_1}{D}\right)^2.$$

The actual revolutions will be—

$$R_d = R_{d1} \text{ (estimated revs., full diam.)} \times \left(\frac{D_1}{D}\right)^{\frac{1}{2}}.$$

Where the engine power forms the basis of the design, this power should be multiplied by $\left(\frac{D_1}{D}\right)^3$ for the power to use in obtaining the full diameter screw.

Such broad tip propellers have one peculiarity, so far as can be ascertained, and that is encountered when they are used on submarines having slips of the first order:

When running on the surface, the revolution correcter is $\left(\frac{D_1}{D}\right)^{\frac{1}{2}}$ as given above. When running submerged and plotting below curve B, Fig. 7, this factor changes to $\left(\frac{D_1}{D}\right)^{\frac{1}{3}}$

The criticism has been made that while the charts will produce correct results for the actual ship, a propeller corresponding to the model propellers used in the model tank can not be produced by the use of the charts of design. It has also been stated that attempts have been made to produce such propellers and the resultant propellers have had blades of extremely narrow form, in fact so narrow as to be not practical.

The answer to this criticism is a very simple one. The critic did not know how to handle the charts to obtain the model propeller. He evidently entered them with the same load fraction, $\frac{e.\,h.\,p.}{E.\,H.\,P.}$, and the same speed fraction, $\frac{v}{V}$, as were used in designing the original propellers for the ship. This would result, when the projected area ratio of the model screw was kept equal to that of the original propeller, in a propeller of less diameter than the model propeller, of very much lower pitch and of exceedingly high revolutions. Should the diameter be held equal to that of the model screw, the pitch obtained will be much lower than that of the model and the projected area ratio much smaller.

Should the law of comparison be applied to the factors used in designing the propeller for the ship, assuming that wake remains constant and that the frictional resistance of the hull follows the law of comparison, which it does not, the proper factors for use in designing the model propeller would be obtained as for example:—

	Ship.	Ratio.	Model
Length on load water line.....L.	L. W. L.	ρ	$\frac{\text{L. L. W. L.}}{\rho}$
Speed of ship.....	v	$\frac{1}{\rho^{\frac{1}{2}}}$	$\frac{v}{\rho^{\frac{1}{2}}}$
Effective horsepower for v.....	$e.\,h.\,p.$	$\frac{1}{\rho^{\frac{7}{2}}}$	$\frac{e.\,h.\,p.}{\rho^{\frac{7}{2}}}$
Basic effective horsepower.....	E. H. P.	$\frac{1}{\rho^{3}}$	$\frac{\text{E. H. P.}}{\rho^{3}}$
Basic horsepower (Engine or Shaft)	H. P.	$\frac{1}{\rho^{3}}$	$\frac{\text{H. P.}}{\rho^{3}}$
Load fraction.....	$\frac{e.\,h.\,p.}{E.\,H.\,P.}$	$\frac{1}{\rho^{\frac{1}{2}}}$	$\frac{e.\,h.\,p.}{\rho^{\frac{1}{2}} \times \text{E. H. P.}}$
Speed fraction.....	$\frac{v}{V}$	$\frac{1}{\rho^{\frac{1}{2}}}$	$\frac{v}{\rho^{\frac{1}{2}} V}$
Basic Apparent Slip.....	S	1	S
Diameter of Propeller.....	D	$\frac{1}{\rho}$	$\frac{D}{\rho}$
Pitch of Propeller.....	P	$\frac{1}{\rho}$	$\frac{P}{\rho}$
Projected Area Ratio of Propeller..	$\frac{P.\,A.}{D.\,A.}$	1	$\frac{P.\,A.}{D.\,A.}$

Taking these values in the equations for power; for ship

(1) $\text{Log H. P.}_d = \text{Log I. H. P.} - Z_{pg}$,

for model

$\text{Log } h.\,p._d = \text{Log } h.\,p. - Z_{pg}$

$Z_{pg} = 1.0414 \text{ Log } \left(\frac{\text{E. H. P.}}{e.\,h.\,p.}\right)$ for ship

$$Z_{pg} = 1.0414 \text{ Log } \left(\frac{\text{E. H. P.}}{e.\,h.\,p.} \times \frac{\rho^{\frac{7}{2}}}{\rho^3}\right) = 1.0414 \text{ Log } \left(\frac{\text{E. H. P.}}{e.\,h.\,p.}\right) + 1.0414 \times \tfrac{7}{2} \text{ Log } \rho$$

(2) $\therefore \text{Log } h.\,p._d = \text{Log H. P.} - 2 \text{ Log } \rho - 1.0414 \text{ Log } \left(\frac{\text{E. H. P.}}{e.\,h.\,p.}\right) - 1.5621 \text{ Log } \rho$

(3) $\text{Log H. P.}_d = \text{Log H. P.} - 1.0414 \text{ Log } \left(\frac{\text{E. H. P.}}{e.\,h.\,p.}\right)$

Subtracting (3) from (2).

$\text{Log } h.\,p._d - \text{Log H. P.}_d = -3.5621 \text{ Log } \rho$

$$\text{Log } h.\,p._d = \text{Log H. P.}_d - 3.5621 \text{ Log } \rho \quad \therefore h.\,p._d = \frac{\text{H. P.}_d}{\rho^{3.5621}}$$

Turning now to the apparent slips and revolutions,

For ship $s = S \frac{\text{H. P.}_d \times 10^{Z_S}}{\text{H. P.}}$

For model $s = S \frac{h.\,p._d \times 10^{Z_S}}{h.\,p.}$

For ship $Z_S = 2.861 \log \left(\frac{V}{v}\right) - .0788$

For model $Z_S = 2.861 \log \left(\frac{V}{v} \times \rho^{\frac{1}{2}}\right) - .0788$

$$h.\,p._d = \frac{\text{H. P.}_d}{\rho^{3.5621}}$$

$$h.\,p. = \frac{\text{H. P.}}{\rho^3}$$

(4) $\text{Log } s = \text{Log S} + \text{Log H. P.}_d + 2.861 \log \left(\frac{V}{v}\right) - .0788 - \text{Log H. P.}$

(5) $\text{Log } s = \text{Log S} + \text{Log H. P.}_d - 3.5621 \text{ Log } \rho + 2.861 \log \left(\frac{V}{v}\right) + 2.861 \times \tfrac{1}{2} \text{ Log } \rho - .0788 - \text{Log H. P.} + 2 \text{ Log } \rho.$

Subtracting (4) from (5)

$\text{Log } s - \text{Log } s = -3.5621 \text{ Log } \rho + 1.4305 \text{ Log } \rho + 2 \text{ Log } \rho.$

$\text{Log } s = \text{Log } s - .1316 \text{ Log } \rho.$

$$s = \frac{s}{\rho^{.1316}}$$

For ship $R_d = \frac{v \times 101.33}{P \times (1-s)}$

For model $r_d = \frac{v \times 101.33}{\rho^{\frac{1}{2}} \times \frac{P}{R} \times \left(1 - \frac{s}{\rho^{.1316}}\right)} = \frac{\rho^{.6316}(v \times 101.33)}{P \times (\rho^{.1316} - s)}$

from which by assuming any desired value of ρ and the known value of s from trials of the ship, the revolutions for the model can be obtained or *vice-versa*.

$$r_d = \frac{R_d \times \rho^{.6316}(1-s)}{\rho^{.1316} - s}$$

To obtain the apparent slip and revolutions of the actual propeller from the performance of the model, the equations become $s = s R^{.1316}$ and

$$R_d = \frac{v \times 101.33}{P(1 - s\,\rho^{.1316})}$$

$$r_d = \frac{v \times 101.33}{\rho^{\frac{1}{2}} \times \frac{P}{R} \times (1-s)} = \frac{\rho^{\frac{1}{2}}(v \times 101.33)}{P \times (1-s)}$$

$$R_d = r_d \frac{(1-s)}{\rho^{\frac{1}{2}}(1 - s\,\rho^{.1316})},$$

depending for its value on ρ and s.

The actual powers and revolutions for the model will probably be higher than those obtained by these equations as the actual effective horse-power is greater than that obtained by the law of comparison and the wake of the model is probably different from that of the ship.

To illustrate the above take first the vessel in the first column of Problem 1 and deduce the model propeller and its performance from the actual by the above formulas, then analyze and estimate the performance of the model screw by the ordinary method.

Conditions of Problem:

Speed of ship = 11 knots,
e. h. p. = 1160
ρ = 30

Analysis.	Comparison.	Analysis.
D = 16′.583	d = ′.5528	
P = 14′.75	p = ′.492	
P. A. / D. A. = .267	P. A. / D. A. = .267	
T. S. = 6230	T. S. = 6230	
1 − S = .86	1 − S = .86	
V = 14.97	V = 14.97	

I. T. $_D = 3.09$	I. T. $_D = 3.09$	
I. H. P. $= 5137$	i. h. p. $= \frac{5137}{\rho^{\frac{2}{3}}} = 5.708$	5.7072
P. C. $= .696$	P. C. $= .696$	.696
E. H. P. $= 3575$	$E.H.P. = 3.9722$	3.9722
S. H. P. $= 4726$	$s.h.p. = 5.251$	5.2506
e. h. p. $= 1160$	e. h. p. $= \frac{1160}{\rho^{\frac{7}{3}}} = .007844$	.007844
$\frac{\text{e. h. p.}}{\text{E. H. P.}} = .3244$	$\frac{\text{e. h. p.}}{E.H.P.} = \frac{.3244}{\rho^{\frac{5}{3}}} = .001974$	.00197475
$Z_p = 1.0414 \text{ Log } \frac{\text{E. H. P.}}{e.h.p.} = Z_p = 1.0414 \text{ Log } \left(\frac{E.H.P.}{\text{e. h. p.}} \times \rho^{\frac{1}{3}}\right) = 2.8164$		
K $= 1.175$	K $= 1.175$	1.175
S. H. P.$_d = 1716$	$s.h.p._d = \frac{1716}{\rho^{3.1671}} = .0093944$	.0094156
V $= 11$	$v = \frac{11}{\rho^{\frac{1}{3}}} = 2.0083$	2.0083
$\frac{V}{V} = .7348$	$\frac{v}{V} = \frac{.7348}{\rho^{\frac{1}{3}}} = .13415$	.13417
$Z_S = 2.861 \text{ Log } \left(\frac{V}{v}\right) -$		
$.0788 = .30409$	$Z_S = 2.4170$	2.4170
$s = .1024$	$s = .065442$	.06538
$R_d = 84.19$	$R_d = 84.19 \frac{\rho^{.6816}(1-s)}{\rho^{.1816} - s} = 442.9$	442.96

Attention is called to the extremely low load and power fractions under which the model screw operates at comparative speeds and loads. Commander McEntee's estimates for the ship are 1750 S. H. P.$_d$ and 84.5 revs.

Now take another case, beginning with the model screw and estimate the power and revolutions for the ship by comparison and by analysis of the actual propeller.

Let us take the case of vessel H.

Speed of ship $= 21$ knots	Diam. of Prop. $= 18'.25$
e. h. p. $= 15575$	Pitch " " $= 19'.75$
Propellers $= 2$ three-bladed.	$\frac{\text{P. A.}}{\text{D. A.}}$ " " $= .328$
$\rho = 30$	

SHEET NO.17
SCALE OF GREATEST IMMERSED BEAM÷LENGTH ON LOAD WATER LINE = B/L.W.L.
SCALE OF H÷B CONDITION 1 AND 3
SCALE OF H÷B CONDITION 2
SCALE OF BLOCK COEFFICI

CHART FOR CORRECTION OF S.H.P. FOR VARIATION IN $\frac{H}{B}$ FROM STANDARD AND FOR BLOCK COEFFICIENT WHERE L.W.L. OF ANY VESSEL DIFFERS FROM STANDARD L.W.L.

E OF BLOCK COEFFICIENTS FOR INNER SCREWS OF 4 SHAFTS
RANGEMENTS ALSO FOR CENTER SHAFT AND FOR SINGLE
REW, THE DEAD WOOD BEING CUT AWAY IN BOTH
LATTER CASES

FIG. 1

SCALE OF COEFFICIENT OF IMMERSED

COEFFICIENT

CURVES OF B.C. ON A-B SHEET NO. 17

SCALE OF H÷B

.7
.6
.5
.4

STANDARD H÷B-HULLS OF TYPE 3

CURVES OF B.C. ON C.-D SHEET NO. 17

SCALE OF H÷B

.7
.6
.5
.4

STANDARD H÷B-HULLS OF TYPES 1 AND 2

.1 .2 .3 .4 .5 .6

SCALE OF BLOCK COEFFICIENT

SHEET NO. 17B

FIG. 1A

CURVES FOR CORRECTION OF BLOCK COEFFICIENT WHEN H÷B IS GREATER THAN STANDARD

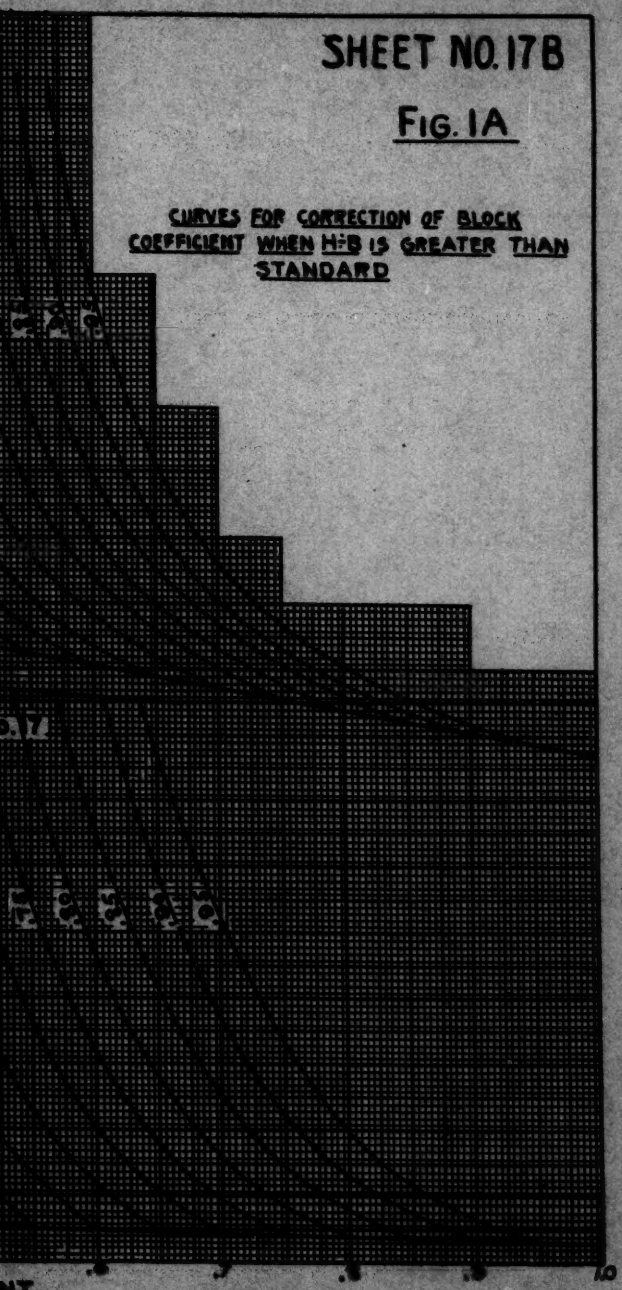
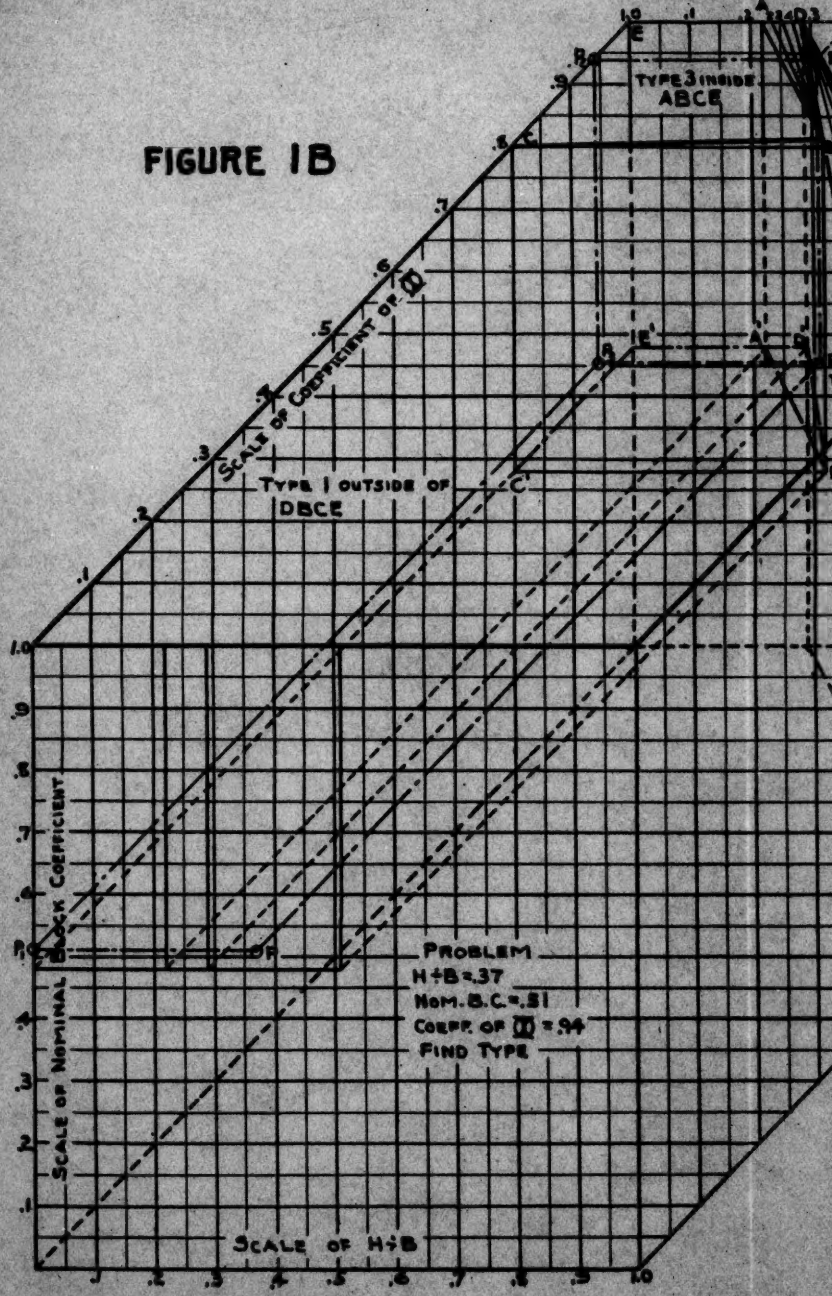
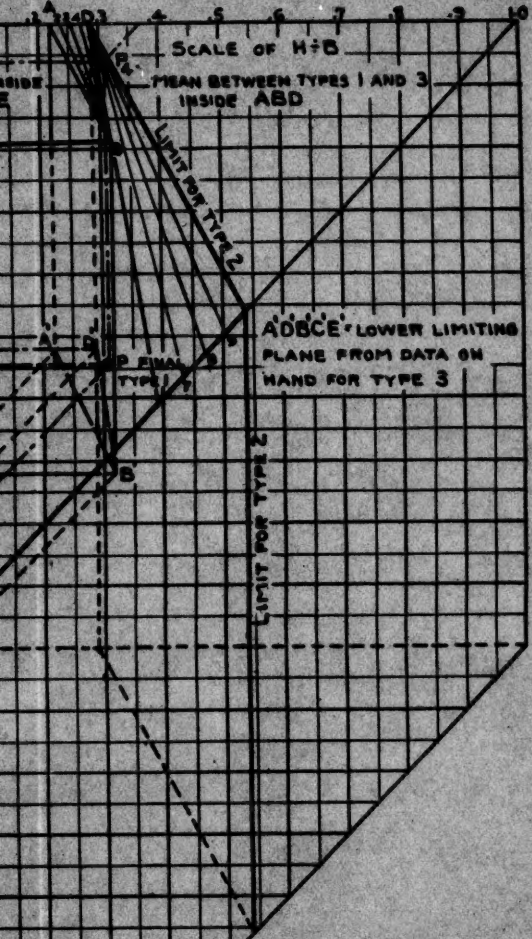


FIGURE 1B


TYPE 3 INSIDE ABCE
SCALE OF COEFFICIENT OF ☒
TYPE 1 OUTSIDE OF DBCE
SCALE OF NOMINAL BLOCK COEFFICIENT
SCALE OF H÷B
PROBLEM
H÷B=.37
NOM. B.C.=.81
COEFF. OF ☒ = .94
FIND TYPE


SCALE OF H÷B
MEAN BETWEEN TYPES 1 AND 3 INSIDE ABD
LIMIT FOR TYPE 2
A'DBCE' = LOWER LIMITING PLANE FROM DATA ON HAND FOR TYPE 3
LIMIT FOR TYPE 2

FORM PRISM
FOR
ESTIMATING HULL TYPES

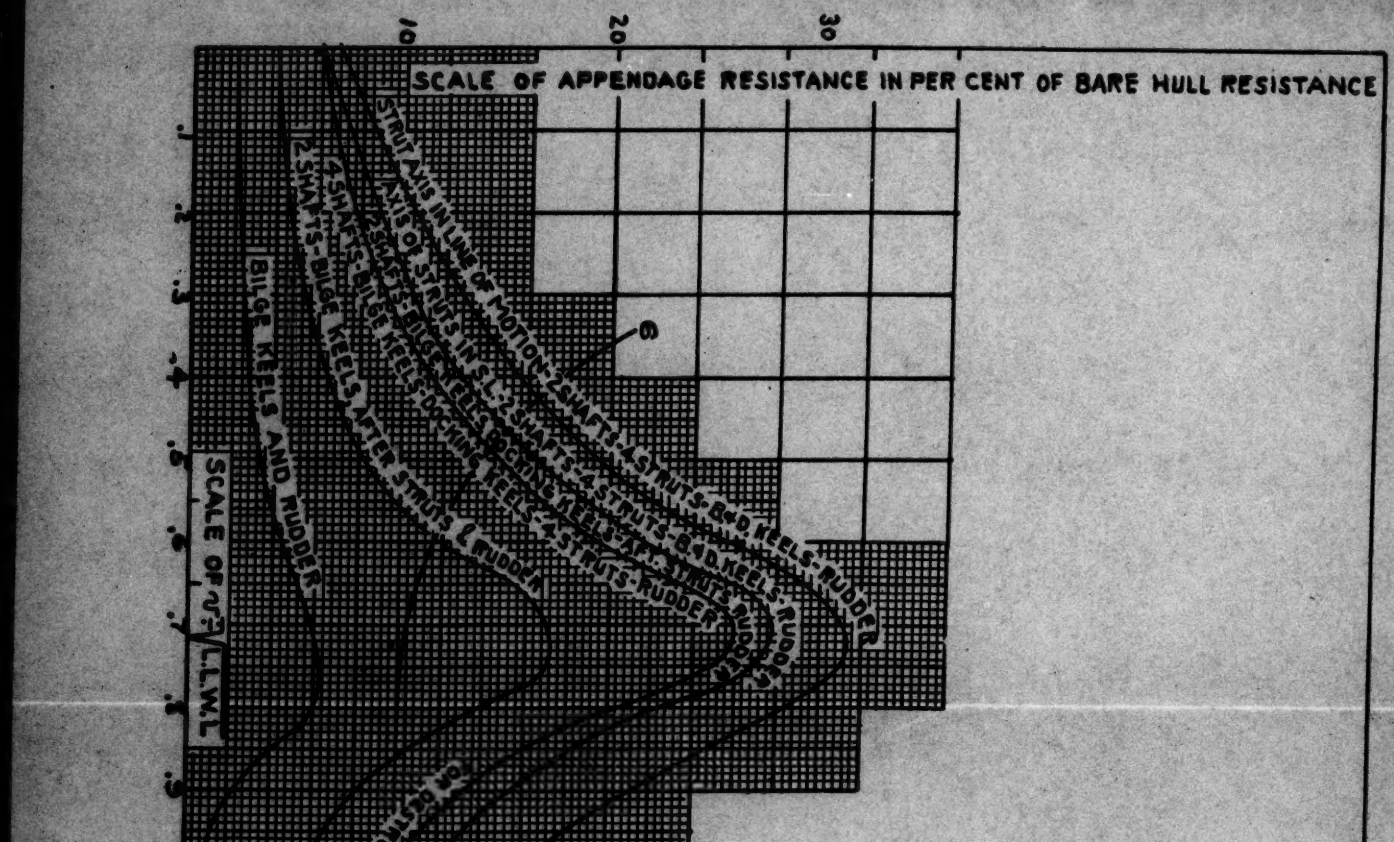
In ABCE-A'B'C'E'; $K = K_3$; $S = S_3$ ..... Type 3

On B2; $K = (5K_3 + 3K_1) \div 8$; $S = (5S_3 + 3S_1) \div 8$
" B3; $K = (K_1 + K_3) \div 2$; $S = (S_1 + S_3) \div 2$ ..... } Type 1&3
" B4; $K = (5K_1 + 3K_3) \div 8$; $S = (5S_1 + 3S_3) \div 8$

" BD; $K = K_1$; $S = S_1$ .....
" D7; $K = K_1$; $S = (5S_1 + 3S_2) \div 8$ .....
" D8; $K = K_1$; $S = (S_1 + S_2) \div 2$ ..... } Type 1.
" D9; $K = K_1$; $S = (5S_2 + 3S_1) \div 8$ .....
" Limit; $K = K_1$; $S = S_2$ .....

" Limit and beyond; $K = K_2$; $S = S_2$ ..... Type 2

All other; $K = K_1$; $S = S_2$ ..... Type 1


10
20
30
SCALE OF APPENDAGE RESISTANCE IN PER CENT OF BARE HULL RESISTANCE
STRUT ARMS IN LINE OF MOTION-2 SHAFTS-4 STRUTS-B&D KEELS-RUDDER
AXES OF STRUTS IN S.S. 2 SHAFTS-4 STRUTS-B&D KEELS-RUDDER
2 SHAFTS-BILGE KEELS-4 STRUTS-B&D KEELS-AFT STRUTS-RUDDER
2 SHAFTS-BILGE KEELS-OPEN KEELS-4 STRUTS-RUDDER
4 SHAFTS-BILGE KEELS AFTER STRUTS & RUDDER
2 SHAFTS-BILGE KEELS AND RUDDER
SCALE OF L.W.L.
B

SHEET NO. 18

FIG. 2

CURVES OF APPENDAGE RESISTANCE

1, 2, 3, 4, 5 - CRUISER STERN SHIPS - TYPES 1 & 3

6 - FANTAIL STERN, TWO SHAFTS, ONE STRUT PER SHAFT, BILGE KEELS - TYPE 2.

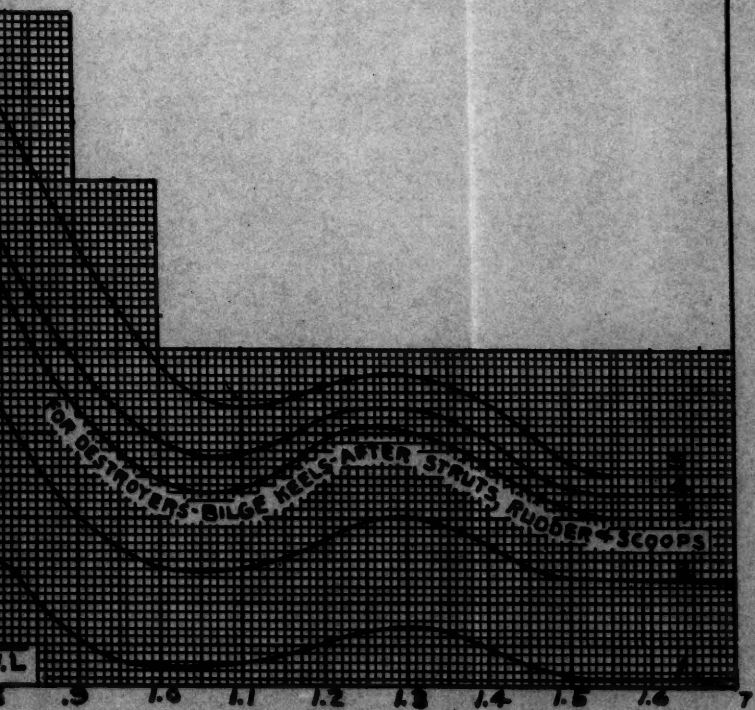


FIG. 3

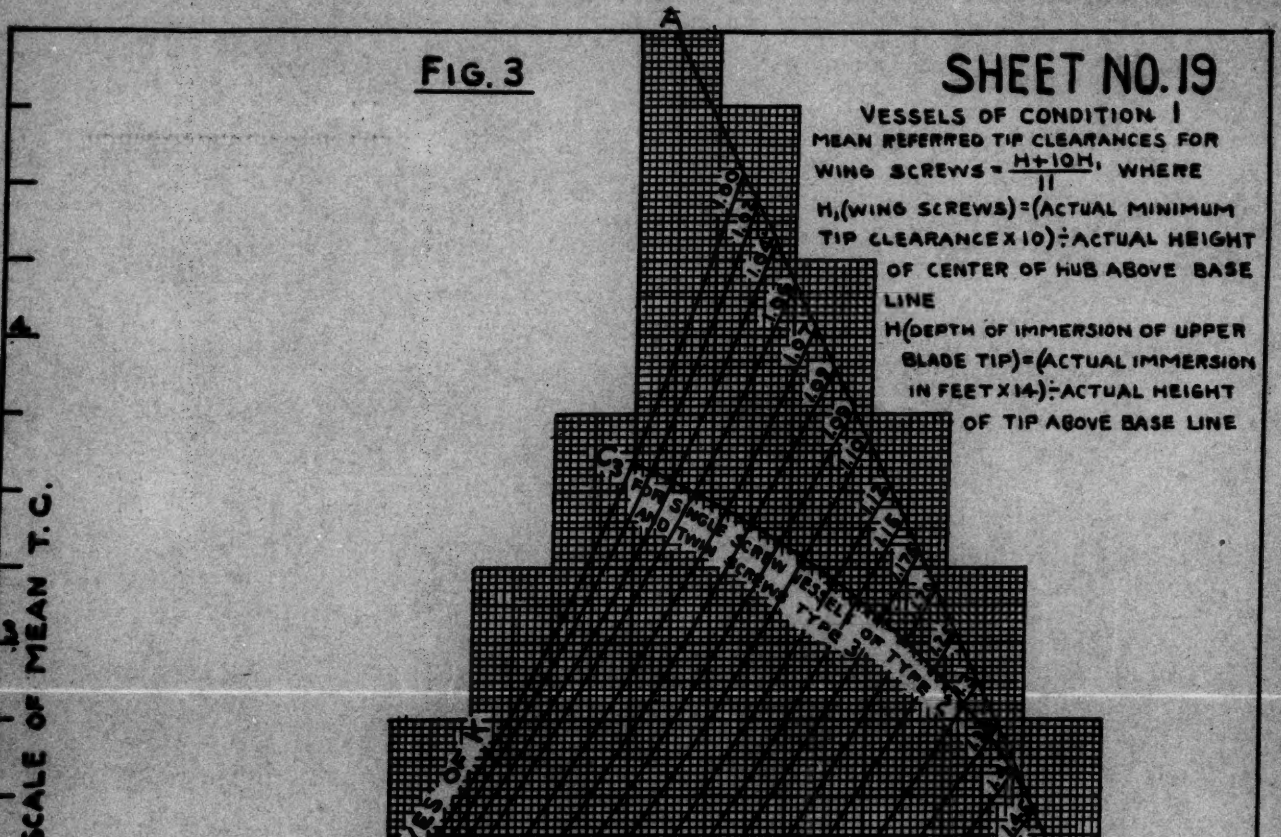
SHEET NO. 19

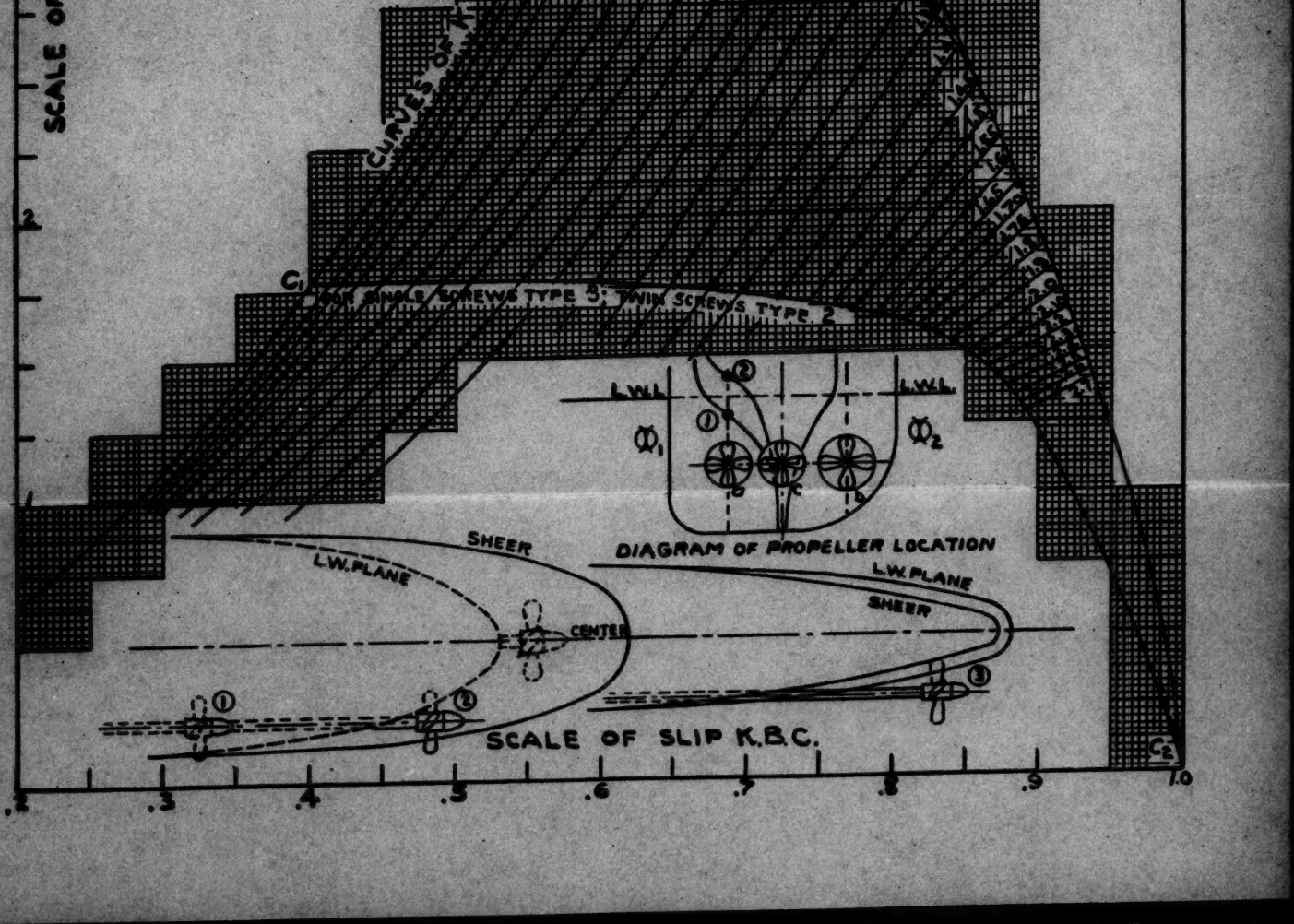
VESSELS OF CONDITION 1

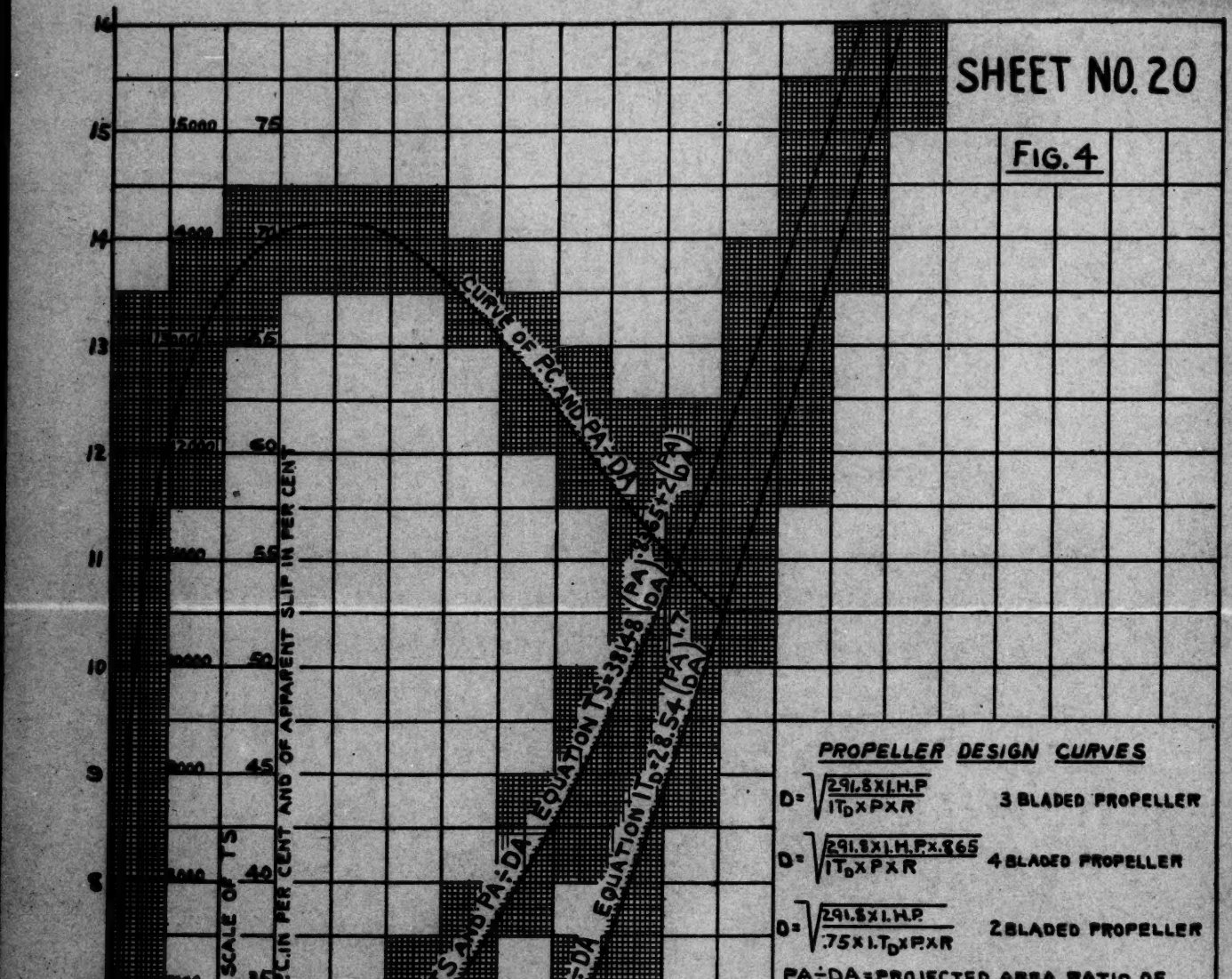
MEAN REFERRED TIP CLEARANCES FOR WING SCREWS = $\frac{H+10H_1}{11}$, WHERE

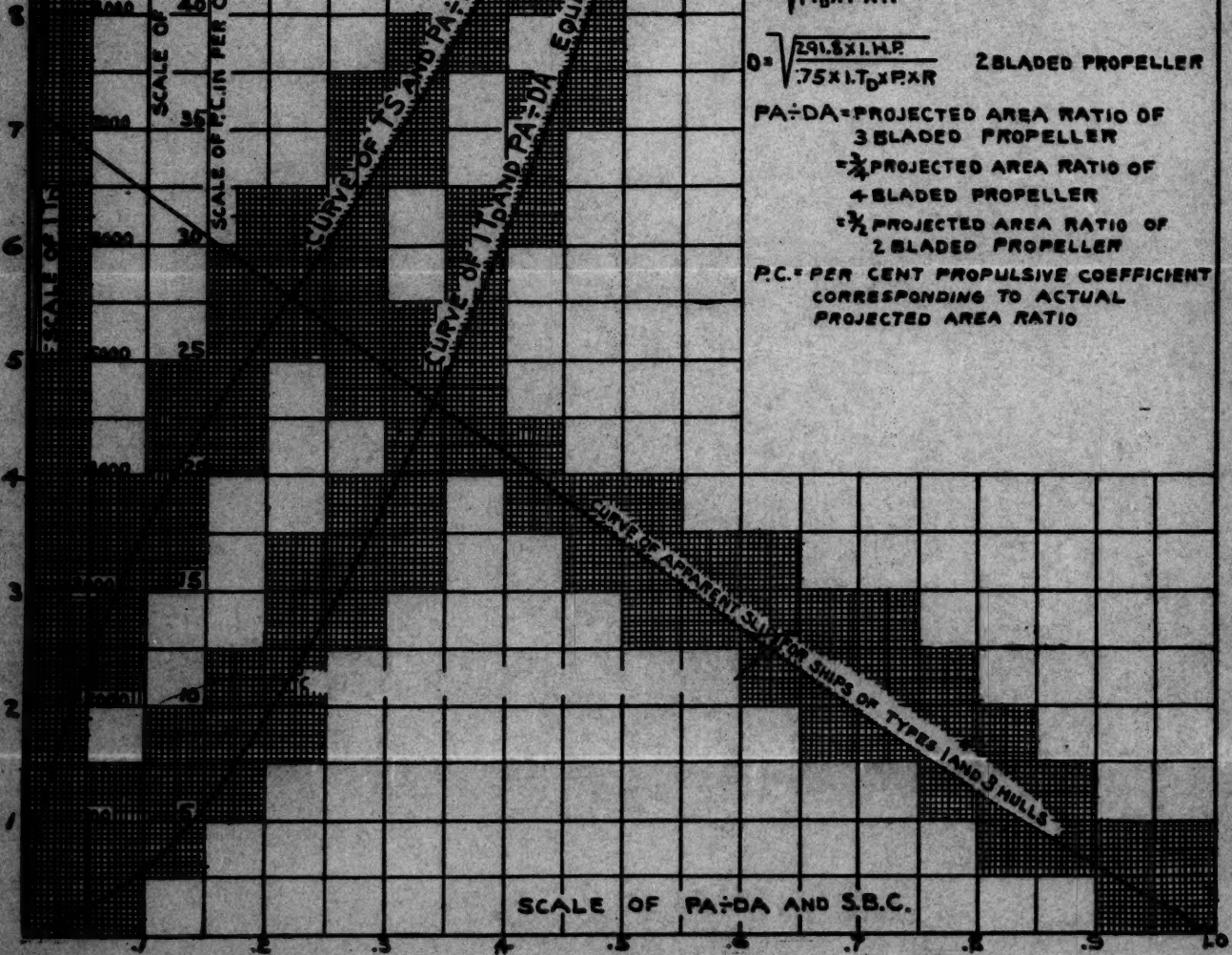
H_1 (WING SCREWS) = (ACTUAL MINIMUM TIP CLEARANCE X 10) ÷ ACTUAL HEIGHT OF CENTER OF HUB ABOVE BASE LINE

H (DEPTH OF IMMERSION OF UPPER BLADE TIP) = (ACTUAL IMMERSION IN FEET X 14) ÷ ACTUAL HEIGHT OF TIP ABOVE BASE LINE




SCALE OF
CURVES OF
C1
SINGLE SCREWS TYPE 3; TWIN SCREWS TYPE 2
L.W.L.
L.W.L.
SHEER
L.W. PLANE
CENTER
DIAGRAM OF PROPELLER LOCATION
L.W. PLANE
SHEER
SCALE OF SLIP K.B.C.
C2
.2
.3
.4
.5
.6
.7
.8
.9
1.0


SHEET NO. 20
FIG. 4
CURVE OF P.C. AND PA÷DA
EQUATION TS=38.48 (PA÷DA)-3.65+2 (PA÷DA)
EQUATION TS=28.54 (PA÷DA)-4.7
SCALE OF TS
P.C. IN PER CENT AND OF APPARENT SLIP IN PER CENT
PROPELLER DESIGN CURVES
D=√(291.8×I.H.P / TD×P×R) 3 BLADED PROPELLER
D=√(291.8×I.H.P×.865 / TD×P×R) 4 BLADED PROPELLER
D=√(291.8×I.H.P. / .75×I.TD×P×R) 2 BLADED PROPELLER
PA÷DA=PROJECTED AREA RATIO OF


SCALE OF P.C.
SCALE OF P.C. IN PER CENT
SCALE OF P.C. IN PER CENT
CURVE OF T'S AND PA÷DA
CURVE OF T'S AND PA÷DA EQUIVALENT
CURVE OF APPARENT SLIP FOR SHIPS OF TYPES I AND 3 HULLS
SCALE OF PA÷DA AND S.B.C.
.1
.2
.3
.4
.5
.6
.7
.8
.9
1.0
1
2
3
4
5
6
7
8
15
20
25
30
35
40
45
5
10
PA÷DA = PROJECTED AREA RATIO OF 3 BLADED PROPELLER
= 3/4 PROJECTED AREA RATIO OF 4 BLADED PROPELLER
= 3/2 PROJECTED AREA RATIO OF 2 BLADED PROPELLER
P.C. = PER CENT PROPULSIVE COEFFICIENT CORRESPONDING TO ACTUAL PROJECTED AREA RATIO
2 BLADED PROPELLER

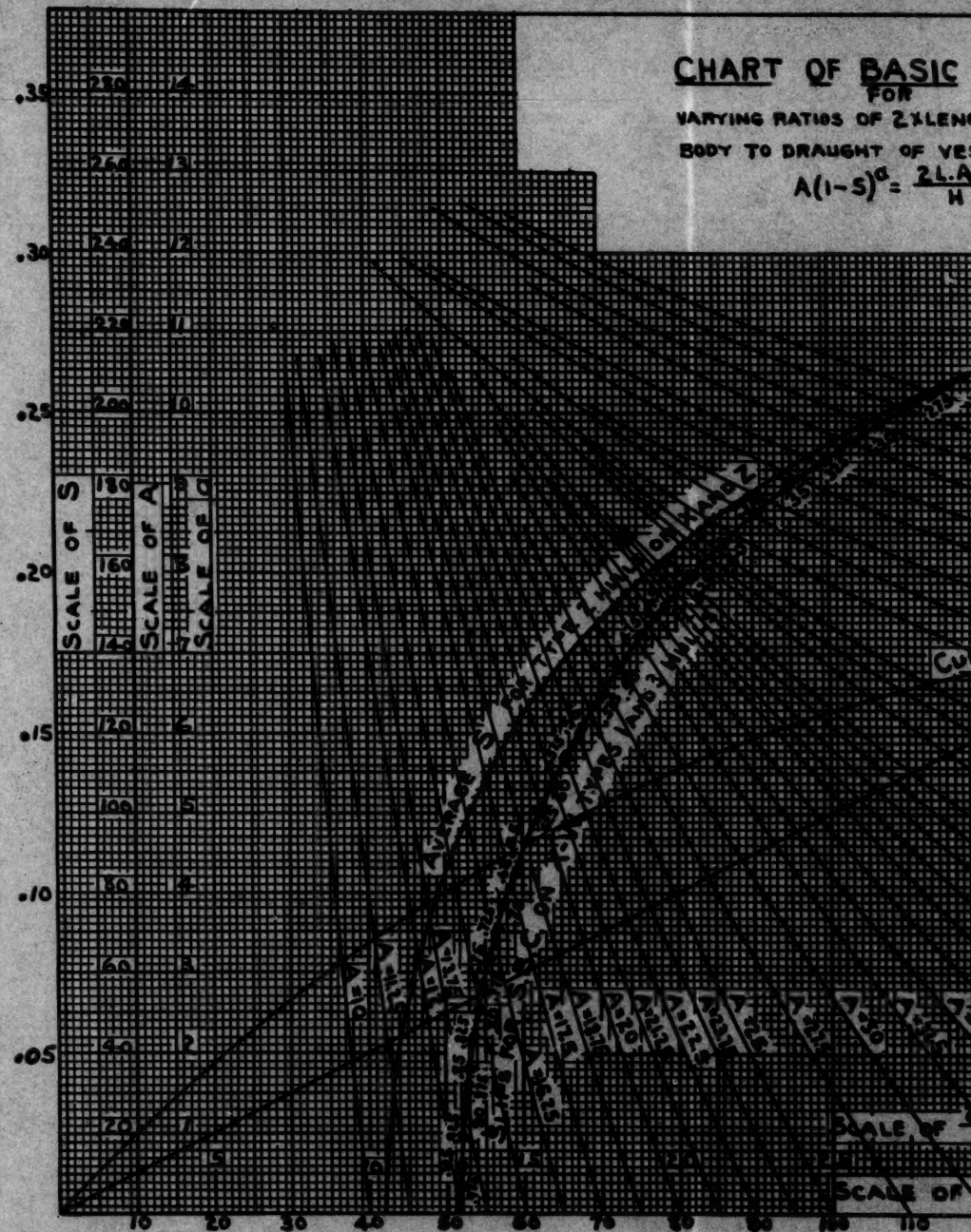
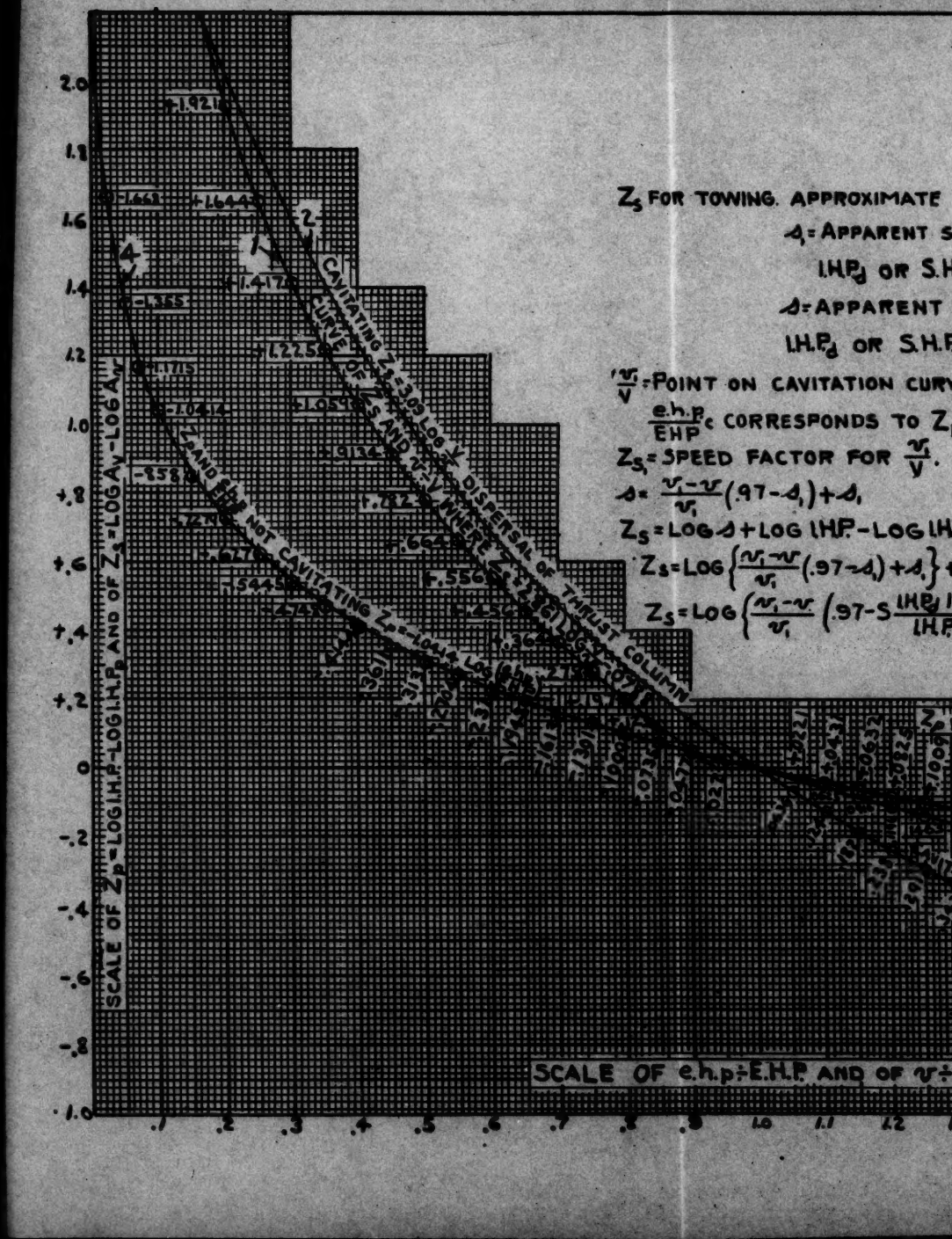


CHART OF BASIC
FOR
VARYING RATIOS OF
BODY TO DRAUGHT OF VE
$A(1-S)^{\alpha} = \frac{2L.A}{H}$
SCALE OF S
SCALE OF A
SCALE OF
.35
.30
.25
.20
.15
.10
.05
10
20
30
40
50
60
70
80
90
100
110

BASIC SLIPS S
FOR
OF 2% LENGTH OF AFTER
HT OF VESSEL
$S)^a = \frac{2L.A.B.}{H}$

SHEET NO. 20-B
FIG. 5


Z_s FOR TOWING. APPROXIMATE
A_1 = APPARENT S
I.H.P$_d$ OR S.H
A = APPARENT
I.H.P$_d$ OR S.H.P
$\frac{v_1}{V}$ = POINT ON CAVITATION CURV
$\frac{e.h.p}{EHP}_c$ CORRESPONDS TO Z_s
Z_{s_1} = SPEED FACTOR FOR $\frac{v_1}{V}$.
$A = \frac{v_1 - v}{v_1}(.97 - A_1) + A_1$
Z_s = LOG A + LOG I.H.P. - LOG I.H
$Z_s = \text{LOG}\left\{\frac{v_1 - v}{v_1}(.97 - A_1) + A_1\right\}$ +
$Z_s = \text{LOG}\left\{\frac{v_1 - v}{v_1}\left(.97 - S\frac{\text{I.H.P}_d}{\text{I.H.P.}}\right.\right.$
CAVITATING $Z_s = 3.05 \log \frac{v}{V}$
CURVE OF Z_s AND $\frac{v}{V}$ WHERE $Z_s = 0$ DISPERSAL OF THRUST COLUMN
AND EHP NOT CAVITATING $Z_p = -10 \log \frac{v}{V}$
SCALE OF Z_p = LOG I.H.P - LOG I.H.P$_d$ AND OF Z_s = LOG A_v - LOG A_{v_1}
SCALE OF e.h.p ÷ E.H.P. AND OF v ÷
2.0
1.8
1.6
1.4
1.2
1.0
+.8
+.6
+.4
+.2
0
-.2
-.4
-.6
-.8
-1.0
.1
.2
.3
.4
.5
.6
.7
.8
.9
1.0
1.1
1.2

SHEET NO. 21

FIG. 6

OXIMATE TYPE 2 SHIPS IN CAVITATION

PPARENT SLIP AT CAVITATION CURVE (FIG. 7) SHEET 22,

P_d OR S.H.P_d CONSTANT

PARENT SLIP BELOW THIS CURVE

OR S.H.P_d = POWER

TION CURVE CORRESPONDING TO I.H.P_d WHERE

IDS TO Z_{pgc} = LOG I.H.P. − LOG I.H.P_d

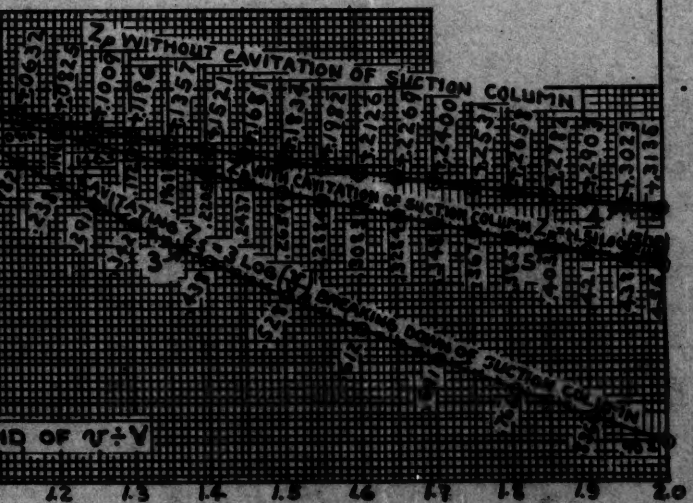
FOR $\frac{n_1}{V}$. Z_s = SPEED FACTOR FOR ACTUAL $\frac{n}{V}$

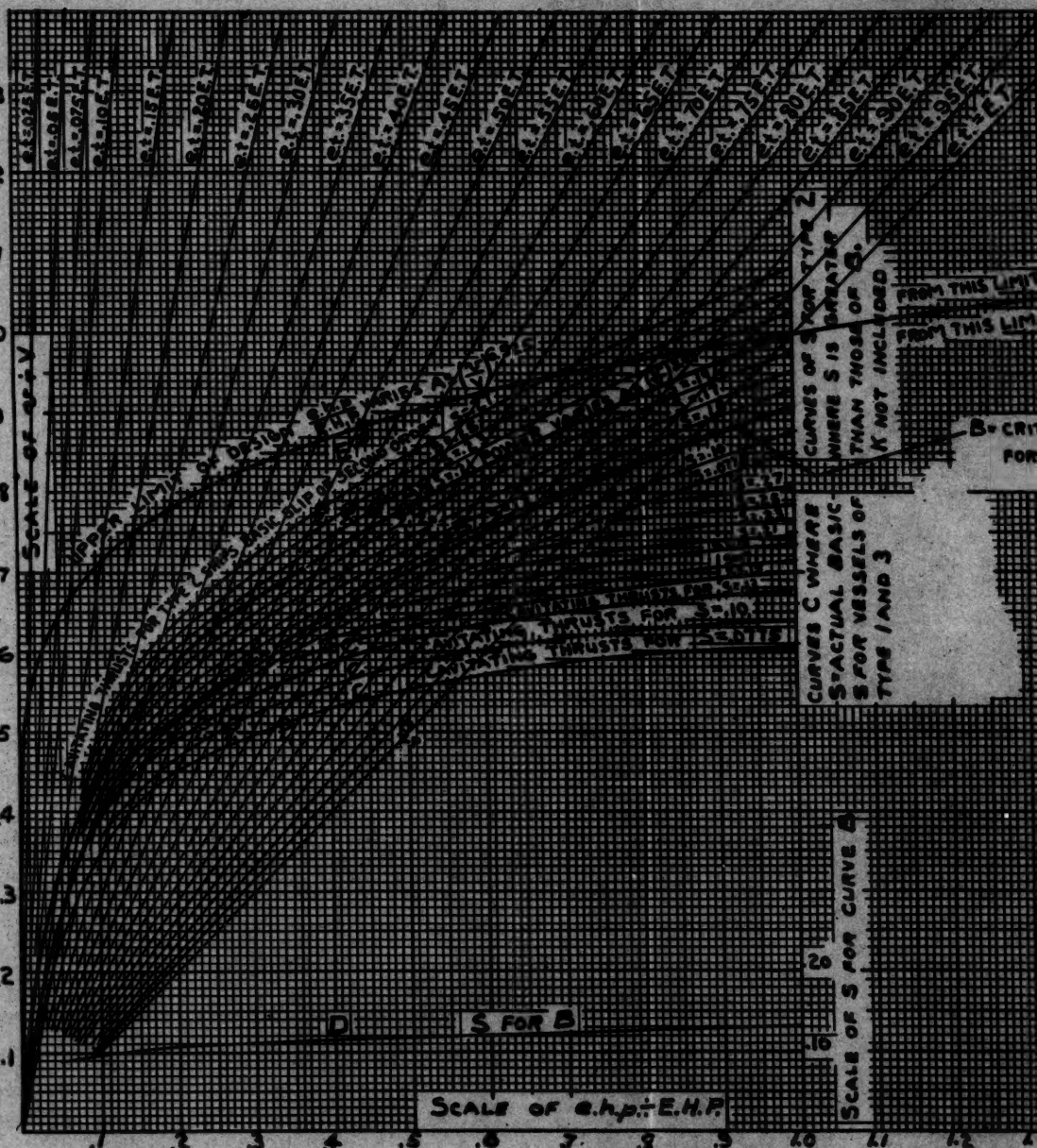
s_1 (1)

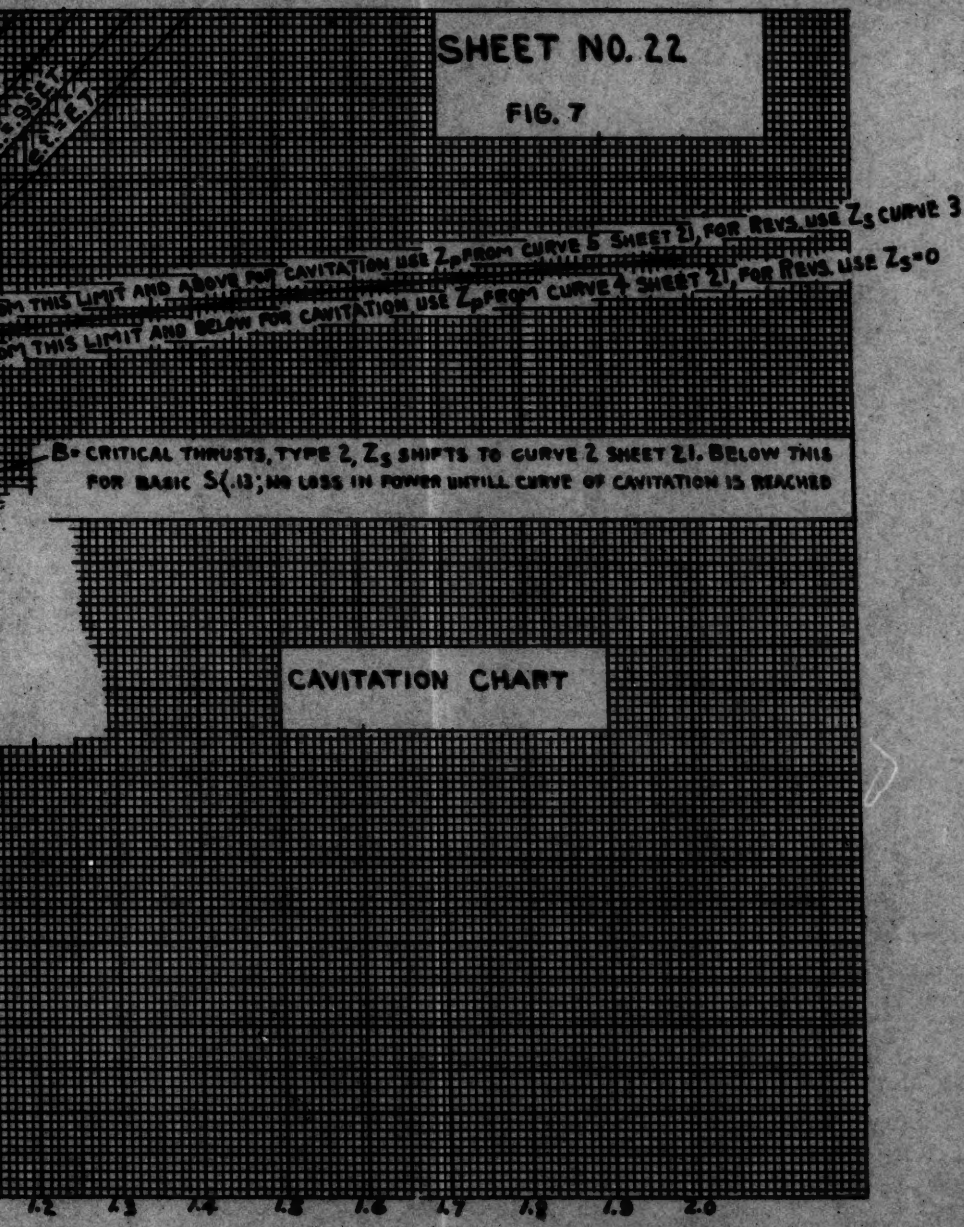
P. − LOG I.H.P_d − LOG S = LOG s_1 + Z_{pgc} − LOG S

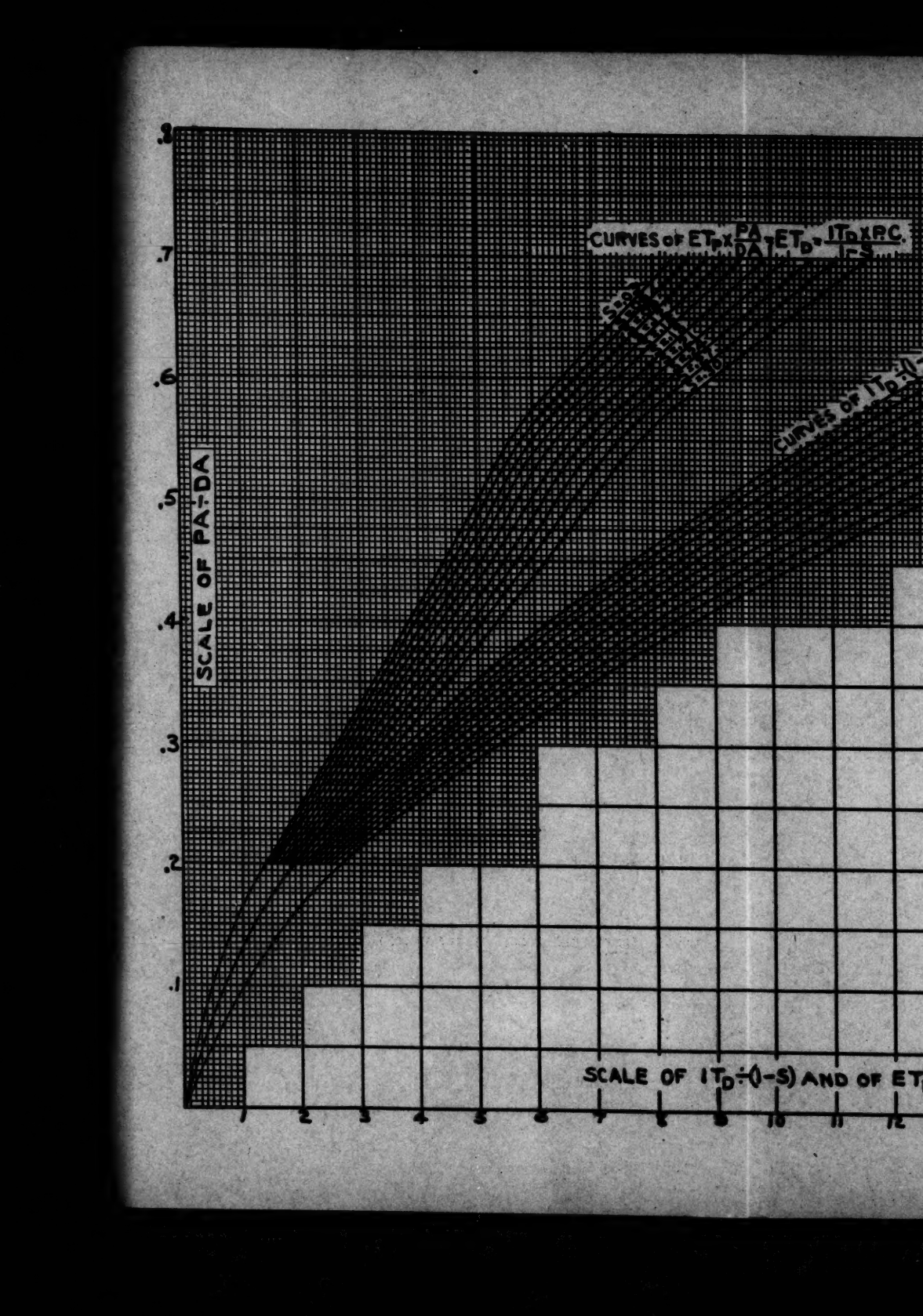
$-s_1) + s_1\Big\} + Z_{pgc} - \text{LOG } S$ (2)

$97 - S\frac{\text{I.H.}P_d\, 10^{Z_{s_1}}}{\text{I.H.P.}}\Big) + S\frac{\text{I.H.}P_d\, 10^{Z_{s_1}}}{\text{I.H.P.}}\Big\} + Z_{pgc} - \text{LOG } S$ (3)




SCALE OF D÷V
SCALE OF e.h.p.÷E.H.P
D
S FOR B
SCALE OF S FOR CURVE A
CURVES C WHERE S=ACTUAL BASIC S FOR VESSELS OF TYPE 1 AND 3
CURVES OF S FOR TYPE 2 WHERE S IS GREATER THAN THOSE OF K NOT INCLUDED
FROM THIS LIMIT
FROM THIS LIMIT
B= CRIT
FOR
1.3
1.2
1.1
1.0
.9
.8
.7
.6
.5
.4
.3
.2
.1
.1
.2
.3
.4
.5
.6
.7
.8
.9
1.0
1.1
1.2


SHEET NO. 22
FIG. 7
OM THIS LIMIT AND ABOVE FOR CAVITATION USE Z_p FROM CURVE 5 SHEET 21, FOR REVS. USE Z_S CURVE 3
OM THIS LIMIT AND BELOW FOR CAVITATION USE Z_p FROM CURVE 4 SHEET 21, FOR REVS. USE Z_S=0
B= CRITICAL THRUSTS, TYPE 2, Z_S SHIFTS TO CURVE 2 SHEET 21. BELOW THIS FOR BASIC S(.13; NO LOSS IN POWER UNTILL CURVE OF CAVITATION IS REACHED
CAVITATION CHART
1.2
1.3
1.4
1.5
1.6
1.7
1.8
1.9
2.0


CURVES OF $ET_D \times \frac{PA}{DA} = ET_D = \frac{IT_D \times RC.}{1-S}$
CURVES OF $IT_D \div (1-$
SCALE OF PA÷DA
.8
.7
.6
.5
.4
.3
.2
.1
SCALE OF $IT_D \div (1-S)$ AND OF ET
1
2
3
4
5
6
7
8
9
10
11
12

SHEET NOS. 23-24

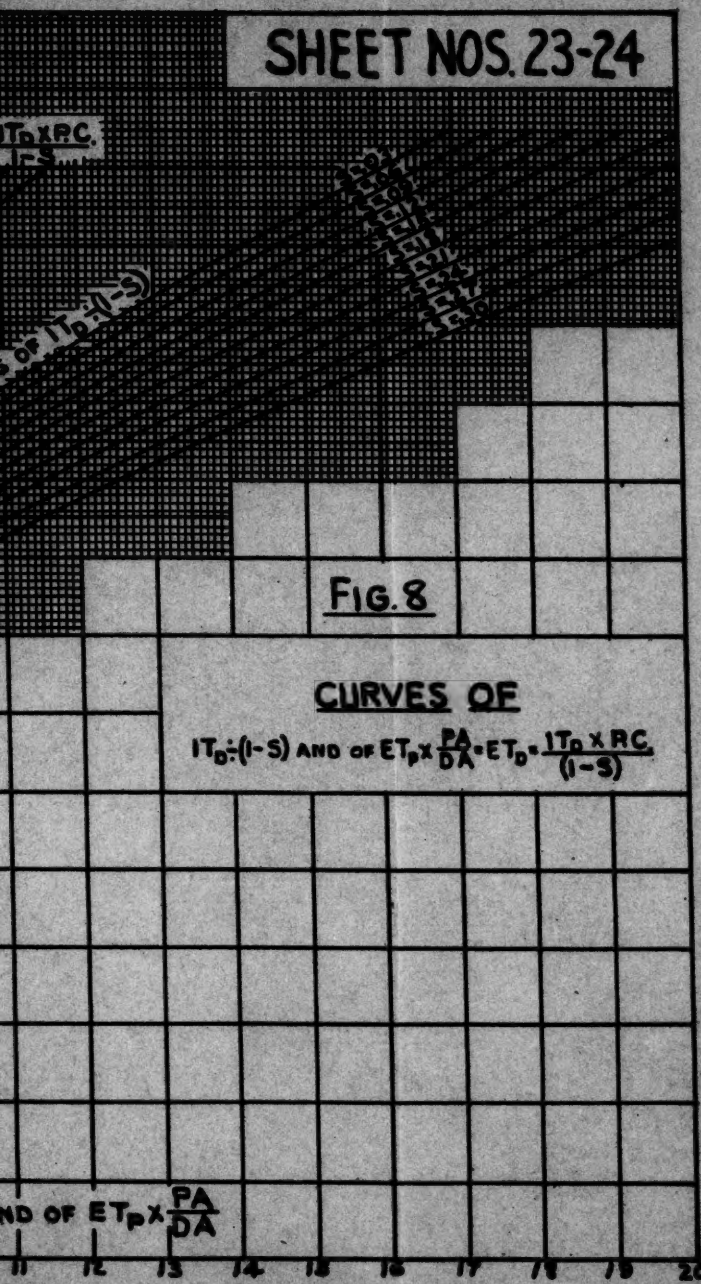


FIG. 8

CURVES OF

$IT_D \div (1-S)$ AND OF $ET_p \times \frac{PA}{DA} = ET_D = \frac{IT_D \times RC.}{(1-S)}$

Data.	Model.	Ship by Comp.	Ship by Analysis	
D	0′ .60833	18′ .25	18′ .25	
P	0′ .65833	19′ .75	19′ .75	
P. A. / D. A.	.328	.328	.328	
T. S.	7250		7250	
1—S	.865		.865	
V	21.32		21.32	
I. T.D	4.35		4 35	
I. H. P.(2 Prop.)	27.557		24800	
P. C.	.665		.665	
E. H. P.	18.3255		16492	
v	3.8341		21	
v / V	.17984		.98504	
e. h. p.	.10593		15575	
e. h. p. / E. H. P.	.0057804		.94441	
Z_p	— 2.3307		— .025868	
K	1	1	1	
I. H. P.$_d$	.128685	23506	23366	Actual 23400
Zs	2.05305		0	
s	.071233		.12719	
R_d	635.41	121.26	123.4	Actual 122.1

The foregoing problems demonstrate that the model screw performance may be estimated either by analysis of the model screw itself or by the application of the laws of comparison.

Now to design the model propeller by fixing the power and speed fractions to be used in the design, take the propeller of the last problem.

D = 18′ .25; P = 19′ .75; P. A. ÷ D. A. = .328; v = 21; e. h. p. = 15575.

$\frac{\text{e. h. p.}}{\text{E. H. P.}} = .94441$; $\frac{v}{V} = .98504$; Power on 2 propellers; S = .135

For the model, the model hull is to be $\frac{1}{10}$ as long on the L. W. L. as the actual vessel

$\therefore \rho = 30$; $d = \frac{18' .25}{30}$; $\frac{v}{V} = \frac{.98504}{\sqrt{30}} = .17984$; $\frac{e.\,h.\,p.}{E.\,H.\,P.} = \frac{.94441}{\sqrt{30^3}} = .0057804$

$e.\,h.\,p. = \frac{15575}{\sqrt{30^3}} = .10593$ $\frac{P.\,A.}{D.\,A.} = .328$ From Fig. 8.

$\frac{e.\ h.\ p.}{E.H.P.} = .0057804$ P. C. = .665

$E.\ H.\ P. = 18.3255$ (2 propellers) $i.\ h.\ p. = 27.557$

$\frac{v}{V} = .17984$ T. S. = 7250

$v = \frac{21}{\sqrt{30}} = 3.8341$ $1 - S = .865$

$V = 21.32$ $P = .65835 = \frac{19' \cdot 75}{30}$

$D = \frac{18.25}{30} = .60833$

$E.\ T._p \times \frac{P.\ A.}{D.\ A.} = 3.3369$

As the model propeller performance can be analyzed by the laws of comparison and as the model propeller can be designed by the use of these laws, it appears that the laws of similitude not only apply to the design charts but also to the propeller itself, and that if such conformity on the part of the charts is to be taken as the proof of their correctness, then judgment in their favor must be given.

When the propeller is working in the range of "cavitation of the thrust column," the estimate of power and revolutions of the actual propeller without cavitation must first be obtained by comparison with the model propeller and the corrections for cavitation applied to these results to obtain the final results; this is necessary as the curves of cavitating thrusts on the chart are not extended down to such low load and speed fractions as those under which the model operates.

CONCLUSION.

The paper as submitted embraces the results of many years of study by the author so far as what may be considered orthodox forms of hulls with orthodox propeller conditions are concerned. The questions of small diameter propellers carried as deeply as possible in the cases of deep draught vessels, propellers for double enders, propellers for tunnel boats, etc., have not been touched. All of these various problems require special study to determine the particular values of K and of S to use in the designs of the propellers.

The problems involved in so recording the results and conclusions of years of study and experience in such a manner that all may readily make use of them have not been easy ones, and even in the final results as submitted there are ranges of indefiniteness which the author, due to lack of reliable data, has no means of clearing up.

Several of these regions of indefiniteness are the determination of the ranges of fineness of hulls covering the transition from hulls of type 2 to those of type 1, and from those of type 1 to those of type 3, also the slight range of speed fraction values at the curve B, Fig. 7, in which the apparent slips pass from slips of the first order to those of the second.

Commander McEntee, with his hull models driven by model propellers, has provided a means by which some of the problems may be solved and by which, when propellers are designed in the safety zones, the true values of K may be obtained, but it is stated that this method of Commander McEntee's is of no use in the cavitation ranges, although the author does not agree with this statement where "Dispersal of the Thrust Column" occurs.

For a complete solution and accurate representation of the phenomena that occur, it is necessary to look to the future, trusting that it will bring with it a vast amount of performance data of sufficient value that it can be used with confidence, and the complete life history of the hull with its propeller plotted in such a manner as to provide an infallible guide to success in the solution of later problems.

The hope has been expressed that in time the true theory of the propeller may be developed from these charts and a composite method, built up from the work of Admiral Taylor and the author, be produced. The author does not hope for this but is of the opinion that his work combined with that of Commander McEntee form a complete system of design and checkage as they stand today, the propellers designed by the author's method to be checked in the model tank by the use of the ship's model and the model of the propeller resulting from reduction in size according to the laws of comparison. This combination fills a long felt want as in the checkage, the model trials give absolute assurance as to the performance of the actual propeller, and enables errors in estimate of K and of S to be corrected, something which has not been had in the past until the actual ship with its propeller have been tried.

It is hoped that the material which has been opened up for inspection will thoroughly impress the student with the fact that the hull with its propeller must be treated in conjunction in the solution of the problem of propulsion. To treat of the propeller without its accompanying hull is to lose time in the ultimate complete solution of the problem involved. The perfect solution of the "Problem of the Hull and its Screw Propeller" rests in the hands of the shipowners, naval architects and marine engineers, and without their hearty and intelligent cooperation in the future, the problem may never be entirely solved.

FUEL OIL IN DIESEL ENGINES.

SOME EXPERIMENTS IN CONNECTION WITH THE INJECTION AND COMBUSTION OF FUEL OIL IN DIESEL ENGINES.*

By Engineer-Commander C. J. Hawkes, R. N. (Ret.), (Member), Professor of Engineering, Armstrong College, Newcastle-upon-Tyne.

The injection of fuel oil in Diesel engines is of the utmost importance, especially in the case of engines of comparatively high speed, and in dealing with this subject references will be made to experiments which have been carried out at the Admiralty Engineering Laboratory in connection with the development of the high-speed engine for naval purposes. The more general method of injecting fuel oil into a Diesel engine is by the use of compressed air, but considerable advance has been made with the, so-called, solid-injection system. It is now generally known that nearly all British submarine engines are fitted with the solid-injection system—and the credit for this development is due to Messrs. Vickers, Limited. With the air-injection system a definite quantity of fuel oil is pumped into the fuel valve casing, and when the fuel valve is lifted off its seat compressed air, provided by a high-pressure air compressor, blows the fuel oil into the cylinder in a very finely-divided state. In the case of the solid-injection system no air compressor is required. The fuel pump raises the oil in the fuel-valve

* Abstract of a paper read before the North-East Coast Institution of Engineers and Shipbuilders, November 20, 1920.

casing, and in the system generally, to a comparatively high pressure, and when the fuel valve is lifted off its seat the requisite quantity of oil is forced, by its own pressure, through a number of small holes in the sprayer—which causes the oil to be split up into a mist.

It is proposed to refer in the first instance to experiments carried out with the solid-injection system and then to certain experiments with the air-injection system. The single-cylinder, four-stroke engine used for the solid-injection tests was the first engine installed in the Admiralty Engineering Laboratory. This engine, which it fitted with a piston of aluminium alloy, has a cylinder diameter of 14½ inches, stroke 15 inches, and develops 100 brake horsepower (nominal) at 380 revolutions per minute. The solid-injection fuel valve used during most of the experiments detailed in this paper was of the direct-lift type, *i.e.*, the fuel-valve spindle, which passed through a gland, was operated directly by the fuel-valve lever. The instant of injection of the fuel was adjusted by means of the timing gear which altered the position of the fuel camshaft in relation to the crankshaft. The fuel pump used during the tests was of bronze and the suction and delivery valves were of hardened steel. The fuel-pump glands were packed with Palmetto packing. Owing to the use of very small holes in the sprayer, it is imperative that the oil should be thoroughly strained, and strainers are consequently fitted on the suction side and also on the discharge side of the fuel pump between the pressure gauge and the fuel valve. The fuel used during all the solid-injection tests referred to in this paper was shale fuel oil having a specific gravity of about 0.86 at 60 degrees F., flash-point 220 degrees F. (close test) and a viscosity of 43 seconds at 70 degrees F. (Redwood No. 1). The power developed by the engine was absorbed by a Heenan and Froude dynamometer. The counter readings and other records were taken every 15 minutes. In some cases only half-hour tests were made, but, even with such short runs, with the apparatus used and with the engine at working temperatures the records were for all practical purposes accurate. The results given are the means of not less than two tests carried out under the same conditions. In all cases, with the exception of a few runs at low powers, the initial, or maximum, pressure was adjusted by means of the timing gear to about 630 pounds per square inch before commencing a test. The compression pressures used during the tests are shown in the tables of results. Throughout the principal tests the jacket, etc., cooling water was kept at a mean temperature of about 80 degrees F.

On account of the situation of the laboratory it was necessary to fit an exhaust pipe about 180 feet in length. The engine exhausted into two silencers fitted with water sprays and consequently steam was produced in the exhaust pipe which made it difficult to ascertain the condition of the exhaust with any degree of accuracy. The drainage water from the first silencer was, however, allowed to pass over a glass plate painted white on the under side, and it was found that by this means the condition of the exhaust could be readily ascertained and the presence of even a small quantity of carbon in the exhaust gases was immediately detected. In the results given, therefore, the condition of the silencer drainage water is recorded.

The aluminum-alloy piston, which was fitted with six narrow cast-iron rings, was lubricated with the same oil as was used for bearing lubrication. The exhaust valve of the experimental engine was water cooled. During tests which were previously made with this engine it was found that the exhaust valve had a tendency to stick even after a comparatively short period of working. When examined it was found

that the clearance between the valve spindle and its guide was always filled with a hard carbon deposit and it was decided to grind the spindle parallel and fit a deflector, as shown in Fig. 1, arranged so that when the exhaust valve opened the spindle was protected from the rush of exhaust gases. This deflector was found to be effective in the experimental engine, and since it has been fitted there has been no sign of sticking of the exhaust valve and the spindle has always been free and clean when removed for examination. The valve spindle is lubricated by means of a mechanical lubricator.

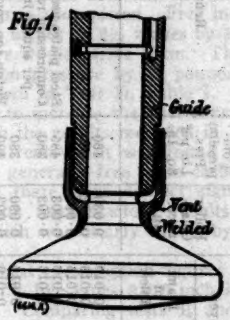


Fig. 1.

Experiments were carried out at the Admiralty Engineering Laboratory with the object of ascertaining the effect of varying the number and size of the holes in the solid-injection sprayer.

Before commencing these experiments, tests were carried out with the single-cylinder engine at 100 brake horsepower and 380 revolutions per minute, with a sprayer provided with five holes 0.019 inch in diameter. This sprayer was similar to the sprayers fitted in submarine engines in 1914. It had previously been found that better results could be obtained by increasing the pressure in the system, and Test A, Table I, was therefore carried out with a fuel pressure of 4,000* pounds, per square inch (gauge). The fuel consumption was 0.45 pound per brake horsepower hour, but it must be remembered that the experimental engine has only a single cylinder and it is fitted with a camshaft driving mechanism which was originally made for a multi-cylinder engine of the same design.

It was thought that slightly better results might be obtained if the fuel sprays were passed over a heated surface and a sprayer was therefore made so that each of the five 0.019-inch holes was surrounded by a steel trumpet expanding from 1/16 inch to 3/16 inch in ¾ inch. But a series of tests made with different modification of the trumpets led to no useful result. It was decided, therefore, to proceed no further in this direction.

Several tests were then made with sprayers provided with holes smaller than the standard 0.019 inch. As there were, at that time, no suitable drills available, the small holes were made by drilling holes about 0.019 inch diameter, caulking them over, and then opening them out again to

* The pressure in the system during all the experiments referred to in this paper was increased as necessary by increasing the clearance between the roller carried by one arm of the fuel valve actuating lever and the fuel cam.

TABLE I.—Results of Tests with Various Sprayers, &c.

Ref. Letter of Tests.	R.P.M.	B.H.P.	Opening of Fuel Valve before t.d.c. Deg.	Fuel Oil.		Fuel Valve.			Compression Pres. Lb. per Sq. In.	Remarks.
				Lb. B.H.P. Hour.	Pres. Lb. per Sq. In.	Sprayer. No. of Holes.	Sprayer. Diam. of Holes.	Roller Clearance. In.		
A	380·0	100·1	20·0	0·45	4,000	5	0·019	0·030	460	
B	381·5	101·0	10·5	0·42	4,200	5	0·016	0·072	460	
C	377·5	99·4	11·5	0·42	4,500	5	0·016	0·095	460	
D	380·0	100·0	19·0	0·47	4,300	5	0·016	0·003	485	Steel plates used.
E	382·9	101·0	23·0	0·43	5,400	5	0·016	0·003	380	Compression reduced to 380 lb. per square inch.
F	380·3	100·3	12·0	0·42	5,600	5	0·016	0·090	380	
G	380·8	100·4	15·0	0·49	5,700	5	0·016	0·003	490	Piston top covered with plates. Compression increased to 490 lb. due to addition of plates.
H	380·3	100·2	8·0	0·46	5,600	5	0·016	0·100	490	Do. do.
J	378·4	99·8	8·5	0·47	5,700	5	0·016	0·100	490	Steel plates. Fuel oil heated to 400 deg. F. on pressure side.

the size required with a fine tapered reamer. The holes were, therefore, 0.019 inch in diameter with a knife-edged orifice of slightly smaller diameter. Fairly good results were obtained with a sprayer having five 0.015 inch holes, but a sprayer provided with five 0.012-inch holes was not promising. The best all-round results were, however, obtained with a sprayer having five 0.016-inch holes, equally spaced. Tests B and C, Table I, give the results obtained with the 0.016-inch hole sprayer. It will be seen that in each case the fuel consumption is appreciably less than with the 0.019-inch hole sprayer.

At this stage it was decided to ascertain the effect of fitting steel plates to the top of the aluminum-alloy piston. Five insulated steel plates were therefore fitted, each 2½ inches by 2½ inches by ⅜ inch thick, in positions such that the fuel sprays would strike them. The fitting of these plates increased the compression pressure by 25 pounds per square inch. The results obtained are shown in Test D, Table I. It will be seen that the effect of the plates was to increase the fuel consumption. Several modifications were tried, with the results recorded in Table I.

When the plates were removed it was found that they had been very hot in places, they were generally free from carbon deposit, and although they were not blistered there were distinct signs of erosion where the fuel jets impinged. The unsatisfactory results were certainly due to delayed combustion—which was clearly shown by the indicator cards and the exhaust temperature.

Up to this time the best consumption figures had been obtained with the sprayer provided with five holes 0.016 inch in diameter, without any special heating devices. Comparing tests E and F, Table I, it will be seen that the fuel consumption was slightly reduced by increasing the fuel-injection pressure. When carrying out tests at higher injection pressures, however, it was noticed that increasing the fuel-valve roller clearance beyond a certain point did not give any better results. It was also noticed that the fuel-valve roller did not follow the contour of the cam, and it was decided to take valve-lift diagrams from the engine when running. For this purpose an extension was fitted to the fuel-valve spindle, and an ordinary indicator was secured to the fuel-valve body so that the spindle extension pressed against the underside of the indicator piston. The drum cord was then connected to an indicator rig which was approximately 90 degrees out of phase with the piston. By this means the fuel valve lift diagrams were obtained. A 200-pound indicator spring was used. Copies of actual diagrams, the ordinates of which give the lift of the fuel valve magnified six times, are shown in Figs. 2 to 5. The quadrant of the fuel valve cut-out lever was notched. Notch 4 represents the position in which the fuel valve had its full lift, notch 2 the lowest lift position at which tests were made and notch 3 an intermediate position.

Fig. 2 shows one of the first valve-lift diagrams taken with the sprayer provided with five holes 0.016 inch in diameter.* The effect of "notching out" on the movement of the fuel valve is clearly indicated. Curves 1, 2 and 3 represent the movement of the fuel valve for each position of the cut-out lever when the engine was barred round, *i.e.*, when the fuel-valve roller followed the contour of the cam. The breaks in these curves, *e.g.*, at a in curve 3, are due to the backlash of the camshaft driving mechanism which is taken up by the load on the fuel-valve spring when the top of the cam passes below the roller. Curves 1, 2 and 3 are, therefore, not correct as drawn, but should be somewhat as shown dotted.

* It was subsequently found that the load on the valve spring in use was slightly less than 618 pounds—which was the load on the standard springs fitted in submarine engines in 1914.

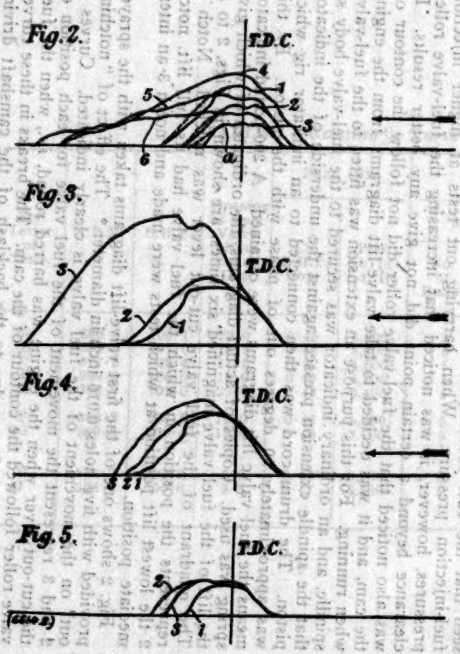


FIG. 2. 5-HOLE SPRAYER. HOLES 0·016, IN. DIAM. REVS. 380. B.H.P. 100.

Curve 1. Barring round Diagram. Notch 4.
Curve 4. Running Diagram, Notch 4. Fuel pressure 3,700 lb.
Curve 2. Barring round Diagram, Notch 3.
Curve 5. Running Diagram, Notch 3. Fuel pressure 5,200 lb.
Curve 3. Barring round Diagram, Notch 2.
Curve 6. Running Diagram, Notch 2. Fuel pressure, 5,600 lb.

FIG. 3. 4-HOLE SPRAYER. HOLES 0·019 IN. DIAM. 4TH NOTCH OF QUADRANT. FUEL VALVE ROLLER CLEARANCE 0.003. REVS. 380.

Curve 1. Barring round Diagram.
Curve 2. Running Diagram. Fuel pressure, 4,100 lb.
Curve 3. Running Diagram. Fuel pressure, 5,000 lb.

FIG. 4. 5-HOLE SPRAYER. HOLES 0·019 IN. DIAM. 4TH NOTCH OF QUADRANT. FUEL VALVE ROLLER CLEARANCE 0·003. REVS. 380.

Curve 1. Barring round Diagram.
Curve 2. Running Diagram. Fuel pressure, 2,500 lb.
Curve 3. Running Diagram. Fuel pressure, 4,000 lb.

FIG. 5. 5-HOLE SPRAYER. HOLES 0·019 IN. DIAM. 2ND NOTCH. ROLLER CLEARANCE 0·090 IN. R.P.M. 380.

Curve 1. Barring Diagram.
Curve 2. Running Diagram. Spring load of 618 lb.
Curve 3. Running Diagram. Spring load of 750 lb.

Valve-lift diagrams were also taken with a sprayer provided with four 0.016-inch diameter holes, and these are shown in Fig. 3. With this sprayer the effect of increasing the fuel pressure is most marked—but it must, of course, be regarded as an extreme case. Valve-lift diagrams obtained with a sprayer provided with five 0.019-inch diameter holes are shown in Fig. 4. Curve 1 is the "barring" diagram, and making the necessary correction for the back-lash in the driving mechanism it will be seen from curve 2 that when the engine was running with the normal fuel pressure of 2,500 pounds, per square inch the roller practically followed the cam. At a pressure of 4,000 pounds per square inch the roller began to leave the cam (curve 3, Fig. 4). In this instance a valve spring was in use loaded to 618 pounds when the valve was in the closed position. The effect of increasing the spring load was next ascertained. Fig. 5 shows the diagrams obtained with the 0.019-inch hole sprayer. Curve 1 shows the "barring" diagram. Curve 2 is the running diagram with a spring load of 618 pounds and a fuel pressure of 4,500 pounds per square inch, and curve 3 is the running diagram with a spring load of 750 pounds and a fuel pressure of 5,200 pounds per square inch. The quantity of fuel oil passed was the same for curves 2 and 3. The effect of the stronger spring is fairly marked but it is still insufficient to cause the roller to follow the cam. Better results were obtained by increasing the load on the spring to 850 pounds, but, with the maximum valve-spindle friction, no appreciable advantage was obtained by increasing the spring load beyond this figure.

It was considered desirable at this stage to investigate the fuel-valve cam, spring, etc., and to ascertain under what conditions "jumping" of the fuel valve was likely to occur. The curves shown in Fig. 6 were therefore prepared. Curve *a* represents the curve of velocity of opening and closing of the fuel valve; curve *b* represents the forces necessary to accelerate and decelerate the valve, etc., curve *c* represents the load due to a 618-pound spring, and curve *d* the spring load available after deducting the forces necessary to accelerate or decelerate the valve and its operating gear.

TABLE II.—VARIATION OF PRESSURE IN INJECTION SYSTEM.

Maximum Pressure. Lb. per Sq. In. (*a*)	Minimum Pressure. Lb. per Sq. In. (*b*)	Fluctuation of Pressure. Lb. per Sq. In.	Mean of (*a*) and (*b*).
4,560	3,440	1,120	4,000
4,320	3,280	1,040	3,800
4,240	3,200	1,040	3,720
4,000	3,040	960	3,520
3,680	2,800	880	3,240
3,480	2,640	840	3,060
3,360	2,560	800	2,960
2,840	2,160	680	2,500
2,360	1,720	640	2,040
2,000	1,400	600	1,700
1,920	1,360	560	1,640

Diagrams had previously been taken from the fuel system with the object of ascertaining the pressure during the cycle. For this purpose an ordinary hydraulic indicator was used, but as whale oil was found

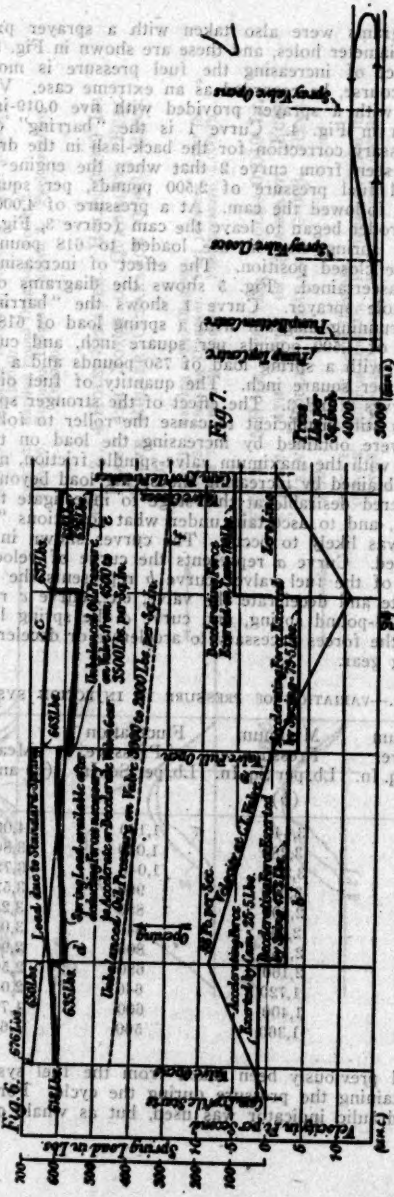


Fig. 7.
Spring Valve Opens
Spring Valve Closes
Fuel Valve Opens
Fuel Valve Closes
Opening
Closing
Lead due to Momentum
Lead due to Acceleration
Lead due to Momentum
Lead due to Acceleration
Spring Load in Lbs.
Velocity in Ft. per Second
Time in Seconds

to leak past the indicator piston a U-pipe connection was made to the indicator which was filled with a viscous oil—so that the heavy oil was in contact with the piston. This overcame the leakage difficulty, and it is considered that reliable records of the variations of pressure were obtained. Table II shows the maximum and minimum pressures recorded when pumping varying quantities of fuel oil, with the engine running at 420 revolutions per minute.

A typical diagram taken from the fuel-injection system is shown in Fig. 7. In this instance the indicator drum was running at engine speed and the fuel pump at camshaft speed. The pump dead centers are marked on the diagram.

Referring again to Fig. 6, curve *e* represents the unbalanced oil pressure on the valve on the assumptions that the full pressure is acting during the opening and closing periods of the valve, and that the pressure drops uniformly in the system from 4,500 pounds to 3,500 pounds per square inch. The former is not, of course, correct for all positions of the fuel valve but the latter is approximately correct. Curve *f* shows a similar curve, but it assumes that the pressure in the system drops uniformly from 3,000 pounds to 2,000 pounds per square inch, giving a mean working pressure of about 2,500 pounds per square inch—which was the pressure for which the system was originally designed. The critical period is undoubtedly when the valve begins to close, and if the full pressure acts on the underside of the valve during this period—and it probably does during the early portion of the closing period—it will be seen that the balance of spring load available to overcome gland friction is practically negligible with the higher fuel pressures assumed. Gland friction is acting against the spring during the closing period, whilst the unbalanced pressure on the valve is always acting against the spring. Fuel-valve gland friction is difficult to estimate in an engine running on service. In the case of the experimental engine it was ascertained that with the fuel-valve gland carefully packed (and tight) a force of 8.5 pounds was required to overcome gland friction (diameter of valve spindle 7/16 inch). This should be regarded as being about the minimum figure. It will be seen that the 618 pounds spring provides only a small margin to overcome an increase in gland friction, or an increased fuel oil pressure beyond that for which the system was originally designed. So far as spindle friction is concerned the use of a "ground" fuel-valve spindle is an advantage—as a gland is then unnecessary. Difficulties following the adoption of higher pressures can be surmounted by fitting stronger springs or by a further modification of the design.

The 0.016-inch hole sprayer used in the tests so far recorded was made, as already mentioned, by caulking over the ends of the holes of a 0.019-inch sprayer and then opening out each hole with a tapered reamer to 0.016-inch. This sprayer showed signs of wear, but it was not excessive, although it had been in use for an appreciable time. Later, suitable drills were obtained and the 0.016-inch holes were drilled in a sprayer of the standard shape. After carrying out a number of tests it was found that the best results were obtained with the 0.016-inch hole sprayer by filing flats in the vicinity of each hole and so reducing the length of each hole to about 1/32 inch.

The success of the solid-injection engine on service, from the point of view of smokeless combustion, depends very largely on the fuel valve and fuel-pump design, correct settings, clean sprayers and tight joints in the system. There should be no difficulty with leakage of connections so long as substantial fittings are provided, but it is advisable to reduce the number of joints to a minimum. It is imperative that every precaution

should be taken to insure the thorough filtration of the fuel oil on its passage to the fuel valve. Although fine filters are fitted occasionally it is found that a choke of one or more holes of a sprayer does occur—especially when the valves and fittings are new. The occasional choking of sprayers on service is probably largely due to pieces of packing working down from the fuel-valve spindle glands, and for this reason, as well as from the point of view of spindle friction, it is advantageous to design the valves so that glands are unnecessary. Pieces of packing from fuel-pump glands also find their way to the sprayers and this points to the desirability of dispensing with packing glands in fuel pumps as well as in the fuel valves.

In the experiments referred to in this paper the best all-round results were obtained with a sprayer provided with five holes 0.016-inch in diameter. Sprayers containing more, and also less, than five holes were tested, but the five-hole sprayer gave the best results within the limits of fuel pressure which the system as designed was capable of withstanding satisfactorily. All the experiments pointed, however, to the desirability, within practical limits, of using smaller holes and higher fuel pressures; and had it been possible to employ higher fuel pressures in the experimental engine there is no doubt that its flexibility would have been increased.

The number and size of the holes in a sprayer depend on the mean pressure in the fuel-injection system; on the viscosity of the fuel oil; on the mean indicated pressure aimed at; on the distance the jets have to penetrate into the combustion chamber and also on the speed of the engine. The angle of the holes, *i.e.*, the angle of the cone on which the holes are drilled, is dependent on the shape of the combustion chamber. So far as the author is aware there are at present no formula which can be used to determine the number, size and angle of holes necessary in a sprayer for a given engine to insure the best results—it is largely a question of trial and error.

Although generally the fuel pump worked satisfactorily during the tests, such pumps should, in the author's opinion, be made from forged steel blocks or steel stampings, when working with fuel pressures of the order mentioned in this paper.

In view of the experience obtained with the direct-lift fuel valve it was decided to design an automatic valve which would completely control the fuel spray, and which would also avoid the necessity of fitting a spindle gland—thus reducing the chances of the valve sticking in the open position or closing sluggishly. In the case of the single-cylinder experimental engine already mentioned it was extremely doubtful whether the existing gears would stand up against its load for any length of time, and with the object of overcoming this difficulty a valve was designed on the lines of that shown in Fig. 8.

With this arrangement the automatic valve is controlled by means of the mechanically-operated spindle or stop A so that the ordinary fuel pump can be used. To operate the valve a lever (not shown in the figure), actuated by the ordinary fuel cam, lifts the spindle A against a spring which corresponds to the spring in the direct-lift type of fuel valve. The fuel pressure in the solid-injection system acting on the piston D then lifts the automatic valve, which is loaded by spring E, and fuel is sprayed into the cylinder. The closing of the automatic valve is brought about either by the fuel pressure falling below that necessary to keep the valve lifted or by the closing movement of the spindle A. It will be seen that the fuel pressure available for spraying is not less than the loading on the valve due to the spring E; and this is a distinct advantage, especially when running at low powers.

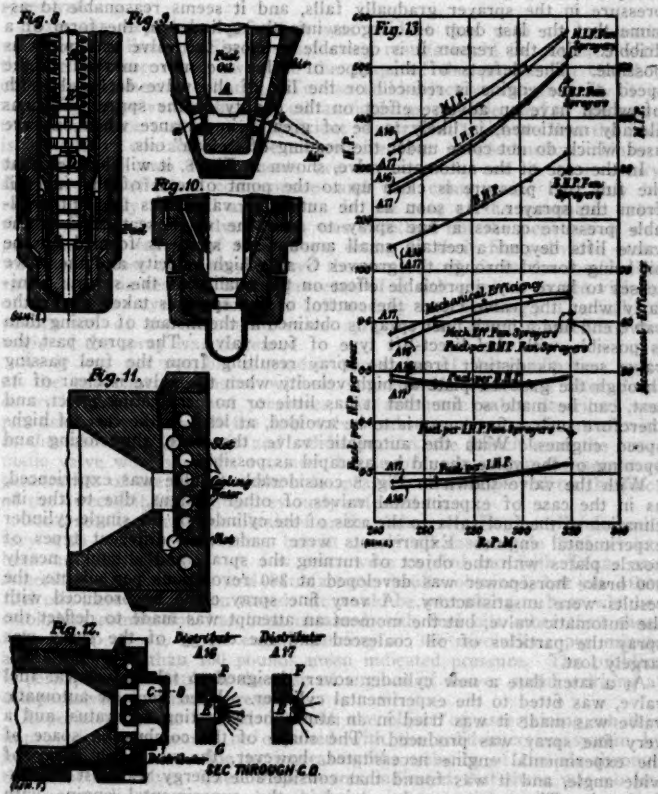


Fig. 8.
Fig. 9.
Fig. 10.
Fig. 11.
Fig. 12.
Fig. 13.
Fuel Out.
Air
Fuel
Cooling Water
Distributor A16
Distributor A17
SEC THROUGH C.D.
B.H.P.
I.H.P.
Mechanical Efficiency
Mech. Eff. Pom. Sprayers
Fuel per B.H.P. Pom. Sprayers
Fuel per I.H.P.
B.P.M.

In the direct-lift type of solid-injection fuel valve the spray is obtained by forcing oil at high velocity through small holes in a sprayer; and the pressure available for producing velocity is controlled by the valve. When the valve commences to lift this pressure, due to wire-drawing past the seat, etc., is less than that available in the system, with the result that the spray at the commencement of injection is probably not particularly good. In a similar manner when the valve closes the pressure in the sprayer gradually falls, and it seems reasonable to assume that the last drop of oil goes into the cylinder in the form of a dribble. For this reason it is desirable to close the valve as rapidly as possible. The defects of this type of valve are more marked as the speed of the engine is reduced or the lift of the valve decreasd—both of which have an adverse effect on the quality of the spray. This, as already mentioned, is likely to be of greater importance when oils are used which do not come under the heading of "clean" oils.

In the case of the automatic valve, shown in Fig. 8, it will be seen that the full fuel pressure is close up to the point of exit of the fuel oil from the sprayer. As soon as the automatic valve lifts the full available pressure causes a fine spray to pass the valve seat. When the valve lifts beyond a certain small amount the spray is formed by the oil being forced through the grooves G at a high velocity and the valve ceases to have any appreciable effect on the quality of the spray. Similarly when the valve closes the control of the spray is taken up by the valve end and a much finer spray is obtained at the instant of closing than is possible with the direct-lift type of fuel valve. The spray past the valve seat, as distinct from the spray resulting from the fuel passing through the grooved plate at high velocity when the valve is clear of its seat, can be made so fine that it has little or no penetrating effect, and therefore such a fine spray is to be avoided, at least in the case of high-speed engines. With the automatic valve, therefore, the closing and opening of the valve should be as rapid as possible.

With the valve shown in Fig. 8 considerable trouble was experienced, as in the case of experimental valves of other designs, due to the inclination of the fuel valve to the axis of the cylinder of the single-cylinder experimental engine. Experiments were made with different types of nozzle plates with the object of turning the spray, and although nearly 100 brake horsepower was developed at 380 revolutions per minute, the results were unsatisfactory. A very fine spray could be produced with the automatic valve, but the moment an attempt was made to deflect the spray the particles of oil coalesced and the energy of the spray was largely lost.

At a later date a new cylinder cover, designed to take a vertical fuel valve, was fitted to the experimental cylinder. When the new automatic valve was made it was tried in an atmospheric testing apparatus, and a very fine spray was produced. The shape of the combustion space of the experimental engine necessitated, however, the use of a spray of wide angle, and it was found that considerable energy was lost in producing it. The valve was then tried in the experimental engine, with various numbers and sizes of grooves in the nozzle plate, but only 66 brake horsepower at 380 revolutions per minute was reached satisfactorily. Better results would doubtless have been obtained had it been possible to use higher fuel pressures; as the energy of the fine sprays produced would thereby have been increased, resulting in an improvement in the penetration, and consequently in the distribution of fuel oil in the combustion space. At low speeds the automatic valve was very satisfactory, and it was found that the engine could be run for long periods at 6 brake horsepower and 116 revolutions per minute, whereas

with the direct-lift type of injection valve it was difficult to run the engine below 25 brake horsepower at 140 revolutions per minute without misfiring. When the automatic valve was in use at the lowest power an indicator diagram was taken with the pencil held on the paper during six successive injections of fuel and only a single line diagram was recorded, indicating that the firing was certain and regular, and that the automatic valve was superior to the direct-lift injection valve for low-speed running.

From the experience gained it was evident that with an automatic valve on the lines of Fig. 8 satisfactory results could be obtained with the fuel pressure used owing to the shape of the combustion chamber of the experimental cylinder. To use solid-injection satisfactorily it is necessary to pulverize the fuel oil as finely as possible, and at the same time to produce a spray with sufficient penetration to insure that the fuel is brought in contact with the air required for its combustion. Unfortunately these two requirements are incompatible, and it is necessary therefore to effect a compromise—and this compromise can only be reached largely by trial and error. In the case of the automatic valve shown in Fig. 8 the pulverization was excellent but the penetration was poor, *i.e.*, the energy of the sprays, owing to their finely-divided state, was rapidly lost when they entered the compressed air in the combustion space, and this resulted in the presence of a rich mixture in the vicinity of the fuel sprayer; which was followed by late burning. The lack of penetration of the automatic valve sprayer was more marked as the angle of the spray was increased.

As it was not possible to modify the shape of the combustion space of the experimental engine without considerable structural alterations, or to employ higher fuel pressures, the automatic valve described was put aside. In nearly all the fuel valves subsequently experimented with at the laboratory, both air and solid-injection, the principle of the automatic valve was, however, retained; but in order to obtain the necessary penetration of the sprays multi-hole sprayers were employed on the lines of the direct-lift fuel valve previously referred to.

Good results have been obtained at the Admiralty Engineering Laboratory with solid-fuel injection, but in nearly all cases shale fuel oil has been used in the tests. A few tests were carried out with Texas and American distillate oils and satisfactory results obtained, but no other fuel oils have so far been tested. It would be interesting to hear whether smokeless combustion has been obtained with fuel oils other than those referred to above, in an engine running at 380 revolutions per minute and at not less than 100 pounds mean indicated pressure. The problem is perhaps less difficult in the case of engines of slower speed working at comparatively low mean pressures.

Experiments were carried out with the object of ascertaining whether improved results would be obtained with the solid-injection system by increased turbulence, due to increasing the velocity of the air through the induction valve. These experiments indicated that there was no improvement due to increased turbulence; in fact, the fuel consumption was slightly increased and the exhaust was slightly shaded. On the face of it, it would appear that turbulence would assist combustion in the case of the Diesel engine—but it will be seen that a high degree of turbulence might have a harmful effect by distorting the spray and causing a portion of it to be brought in contact with the comparatively cool walls or cover of the engine before the oil globules had completely vaporized. This action would produce a smoky exhaust. It is very essential that a fuel spray should not strike a cool surface, and as a high de-

gree of turbulence would have a tendency to bring this about it is considered that turbulence is not so desirable in the Diesel engine as it undoubtedly is in the explosion engine. The turbulence referred to is, of course, that due to high air speeds through the induction valve and not to the use of blast air in air-injection engines. Blast air splits up and distributes the fuel oil in the combustion space, but it acts symmetrically about the center line of the sprays and its direction of action is under control (depending on the position, size and shape of the holes in the distributor or flame plate); consequently it is much less likely to drive a portion of the fuel on to the cool walls of the liner and cover.

There is no doubt that solid injection has much to recommend it, so long as it is in every way satisfactory on service. By its adoption it is not necessary to make provision for an air compressor for supplying blast air—and the high-pressure compressor, as is well-known, has in some instances been a continued source of trouble. Air injection is, however, principally used today, and it seems fair to presume, therefore, that many manufacturers have largely, if not entirely, overcome their compressor troubles. So far as present experience is concerned it is considered that a greater variety of fuel oils can be used satisfactorily with air injection than with solid injection. With the fuel oils experimented with at the laboratory it has been found that up to about 100 pounds mean indicated pressure the fuel consumption with solid injection is approximately the same as the fuel consumption with air injection, at the same power and speed, after taking into account the power required to drive the injection compressor. This means that whilst the consumption per brake horsepower is the same with both systems the consumption per indicated horsepower is less with the air-injection engine than with the solid-injection engine—*i.e.*, the combustion of the fuel oil in the air-injection engine is slightly better than in the solid-injection engine. This difference is no doubt largely due to the better and more rapid distribution of the fuel oil mist in the compression space, which naturally follows the use of air for injection purposes, as the sprays have more energy, and the air when it expands in the combustion space carries the fuel-oil mist with it. Owing to the cooling effect of the injection air, however, a higher compression is necessary in the air injection engine than in the solid-injection engine, for the same fuel oil. It is also found that the solid injection engine starts more readily from the cold condition than the air-injection engine, although in this connection much depends on the compression ratio and on the type of sprayer employed.

At mean indicated pressures at and above 110 pounds per square inch, the experimental engine running at 380 revolutions per minute, smokeless combustion could not be satisfactorily obtained with the solid-injection system, and as it was desirable that higher mean indicated pressures should be developed, it was decided to experiment with the air-injection system. The ordinary air-injection valve will be referred to later, but it is proposed briefly to refer, at this stage, to a valve which may be regarded as a combination of the air and solid-injection systems.

In the ordinary air-injection valve the fuel is forced into the fuel-valve casing against the pressure of the injection air, but no advantage is derived from the pressure to which the fuel oil is subjected. In the modified valve the oil is sprayed in the manner common to solid-injection systems, but inside the cylinder it meets a jet or jets of air so that the distribution is thereby improved. It was hoped that by the use of such a valve a smaller quantity of blast air would be required than is necessary with the ordinary air-injection valve. Further, in the event

of the air compressor failing the engine would be able to run at a slightly lower power satisfactorily on solid injection alone. Various valves were tried based on this principle, both with the air jets surrounding the fuel jets, and also with the fuel jets surrounding the air jets, but it is not possible within the limits of this paper to refer to more than one valve which has given promising results when using shale fuel oil.

Fig. 9 shows diagrammatically a section of the sprayer end of the valve. A is lower end of the automatic valve and the fuel oil passes into the combustion space through the small holes, *a*, *b*, etc., when the valve is lifted. Four air holes surround each fuel exit, so that four small air jets meet the fuel spray in the combustion space at a small angle. In the experimental engine an air valve operated by a separate cam controls the supply of high-pressure air to the air channels shown. This arrangement was made to facilitate experiment, but it is probable that in a valve of this type for service use one cam would be used for controlling the admission of both air and fuel. The engine can be started on the solid-injection system alone, and therefore it is possible to employ a lower compression ratio than would suffice to ensure an easy start from the cold condition with the ordinary air-injection system in use.

It has so far been found that with this valve smokeless combustion can be obtained with a mean indicated pressure of just over 130 pounds per square inch—the experimental engine running at 380 revolutions per minute. The consumption of blast air is slightly less than with the ordinary air-injection valve. The fuel consumption at about 130 pounds mean indicated pressure, using shale fuel oil, is just under 0.41 pound per brake horsepower, but as the air compressor is driven by an electric motor this does not take into account the power required to supply the blast air. The valve combining the solid and air-injection systems has some advantages over the ordinary air-injection valve, but it undoubtedly involves greater complication. Experiments are being continued, and the question of its adoption will, of course, depend upon the results obtained after prolonged trials.

When using the ordinary air-injection system for developing comparatively high mean pressures in high-speed engines it is necessary to use a distributor provided with holes so that a number of small jets are produced. Fig. 10 shows the lower end of a fuel valve designed for a 20-inch by 20-inch experimental four-stroke cycle engine. The number and size of the holes in the distributor to give the best all-round results is a question of trial. The angle of the holes, as in the case of the solid-injection sprayer, is largely dependent on the shape of the compression space.

A few details of test carried out with various fuel-valve distributors in a single-cylinder opposed-piston two-stroke engine may be of interest. This engine was designed and constructed by Messrs. William Doxford and Sons, Sunderland, originally for their own use, but before the engine had been run it was transferred to the Admiralty Engineering Laboratory. The engine follows the firm's usual design and has a cylinder diameter of 14.55 inches and a combined stroke of 28.3 inches. It is fitted with two water-cooled fuel valves. The double-acting scavenge pump and four-stage air compressor are driven from two cranks on an extension of the engine shaft. The scavenge pump and compressor were made larger than necessary for ordinary running, for the purpose of supercharging experiments at a later date, and consequently this reduced slightly the mechanical efficiency of the engine. During the tests referred to in this paper the engine, owing to the lay-out of the laboratory, exhausted into a pipe 180 feet in length.

The sprayer or distributor, of each fuel valve as originally designed was provided with two narrow slits intended to give horizontal fan-shaped sprays. The center line of each spray was about ½ inch from the top of the adjacent piston—the distance between the pistons at the end of the compression being about 1.75 inches. A number of tests at various powers were carried out with the original distributors shown in Fig. 1 with various shapes of cams, etc., and it was found that the fuel consumption at the higher powers and speeds increased rapidly with an increase in mean pressure. From the experience gained with the experimental 14½-inch by 15-inch four-stroke engine it was clear that fuel sprays having a comparatively large surface were not satisfactory, as when working at high speeds and high mean indicated pressures late burning resulted. It was decided, therefore, that further improvements in the fuel consumption of the opposed-piston engine could only be obtained by replacing the original distributors by distributors provided with a number of holes. The design of the original valves was such that it was necessary to cut off the ends of the valve bodies and replace them by ends which could be adapted to take experimental distributors provided with circular holes. Fig. 12 shows the experimental distributors as fitted.

Each distributor was provided with two rows of holes. The center lines of the holes were made parallel to the flat tops of the pistons, and the angles of the holes in the horizontal plane were arranged to give the best distribution. The diameter of each hole depended on its distance from the cylinder liner in the direction of the spray and on the volume of that portion of the combustion space which it had to feed, and it was arranged that each row of holes should sweep over approximately half the surface of the adjacent piston. (The center lines of the two fuel valves in the horizontal plane are parallel but slightly off-set.) A number of distributors of this type were tested, and it was found that the best all-round results were obtained with the distributor A 16, of which a horizontal section is shown in Fig. 12. The upper row of holes in each distributor is indicated in the figure by the full lines and the lower row of holes by the dotted lines. Table III gives the sizes and angles of the holes. The mean results of a series of tests carried out with the A 16 distributors are shown in Table IV and Fig. 13. The fuel used was Texas fuel oil having a specific gravity of 0.91 at 60 degrees F., flash-point 195 degrees F. (close test), viscosity 318 seconds at 70 degrees F. (Redwood No. 1), and a calorific value of 18,400 British thermal units (lower heating value). The exhaust was clear throughout the tests.

The maximum power developed by the A 16 distributors was 406 brake horsepower at 364 revolutions per minute, the mean indicated pressure being 150 pounds per square inch. The exhaust was not black at this power, but it was dark.

The results obtained with the original fan-shaped sprays at 320 revolutions per minute are shown in Fig. 13. It will be seen that the A 16 distributors gave much better results in regard to fuel consumption than the fan sprays.

It was then decided to fit distributors on the lines of distributor A 17, Fig. 12. In this distributor, particulars of which are given in Table III, each row sweeps the whole of the piston surface and the holes are of more uniform size. The mean results of the series of tests with these distributors, shown in Table IV and Fig. 13, do not differ in any marked degree from the results with the A 16 distributor.

TABLE III.

Distributor A 16.

Diameter of hole, in.	0·031	0·047	0·055	0·058	0·062	0·063	
Angle of top jets, deg.	37·0	51·5	63·5	74·0	85·0	92·5	Angles measured from EF.
Angle of bottom jets, deg.	21·0	34·0	46·5	57·5	66·5	76·0	Angles measured from EG.

Distributor A 17.

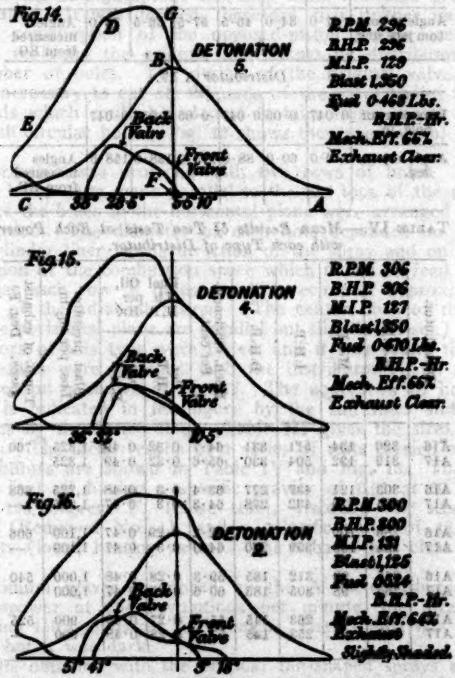
Diameter of hole, in.	0·047	0·05	0·047	0·05	0·058	0·047	
Angle of jets, deg.	40·0	60·0	88·0	115·0	138·0	158·5	Angles measured from EF.

TABLE IV.—*Mean Results of Two Tests at Each Power with each Type of Distributor.*

Distributor.	R.P.M.	M.E.P.	I.H.P	B.H.P.	Mech. efficiency. Per cent.	Fuel Oil, Lb. per H.P.-Hr.		Blast pressure. Lbs. per sq. in.	Temp. of exhaust gases. Deg. F.
						Per I.H.P.	Per B.H.P.		
A16	320	134	571	331	64·7	0·32	0·49	1,325	760
A17	319	132	504	330	65·6	0·32	0·49	1,325	—
A16	303	121	437	277	63·4	0·3	0·48	1,225	668
A17	304	119	432	278	64·3	0·3	0·47	1,225	—
A16	286	109	370	230	62·2	0·29	0·47	1,100	606
A17	285	105	359	230	64·0	0·3	0·47	1,100	—
A16	265	99	312	185	59·3	0·28	0·48	1,000	540
A17	265	98	305	185	60·6	0·29	0·47	1,000	—
A16	247	89	263	145	55·2	0·27	0·49	900	525
A17	247	86	253	145	57·3	0·28	0·48	900	—

Texas fuel oil used, specific gravity 0.91 at 60 degrees F. Exhaust clear throughout. Temperature of exhaust taken by thermocouple. Cams in use: Front cam opened 19 degrees before T.D.C., closed 41 degrees after T. D. C.; back cam opened 6 degrees before T.D.C. and closed 32.5 degrees after T.D.C.

In all the distributors experimented with the mean distance of the rows of holes from the tops of the adjacent pistons when at the end of the compression stroke was 29/64 inch. It was arranged, therefore, to test distributors with the rows of holes about 9/16 inch from the piston, but otherwise similar in all respects to the A 17 distributors. The fuel consumptions at the higher mean pressures were not so good as the consumptions with the A 16 or A 17 distributors. This bears out Professor Junkers' contention in regard to the disposition of the fuel sprays, viz., that when working at high mean pressures with the opposed-piston engine it is necessary to arrange the fuel sprays as close as practicable to the pistons.

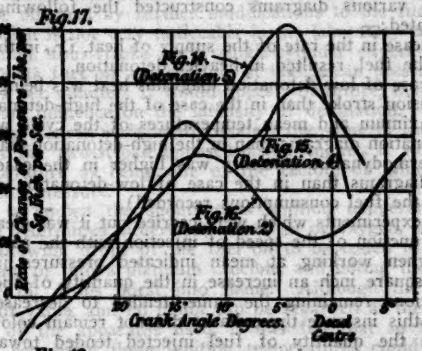


In nearly all engines of the opposed-piston type which the author has seen running the injection of the fuel oil has been accompanied by slight detonation. These engines, however, did not run much above 200 revolutions per minute, whereas the experimental opposed-piston engine, referred to in this paper, was run at speeds up to 360 revolutions per minute. In certain circumstances the detonation of the fuel in the experimental engine was most marked, and although it did not appear to be harmful it would, if heavy, be rather distressing in a multi-cylinder engine of this design.

Three typical combustion diagrams, obtained with various shapes of cams, are shown in Figs. 14 to 16. The valve lift diagrams for the

front and back fuel valves are also shown on the same angle base. An arbitrary figure for the intensity of detonation and other particulars are given against each combustion diagram. Although in the experimental opposed-piston engine there was no separate timing gear, and therefore it was not possible to modify the "timing" of the fuel valves when the engine was running, the fuel cams were adjusted so that the maximum pressure was limited to about 630 pounds per square inch.

In Fig. 14, with short period cams in use, A B C is the diagram obtained when the engine was running at about 300-brake horsepower, but with the fuel momentarily cut off. F A B therefore represents, on an angle base, the compression curve, and B C F the expansion of the air previously compressed. The line F G gives the position of the indicator drum corresponding to the inner dead center, *i.e.*, when the pistons were at the end of the compression stroke. A D E represents the combustion



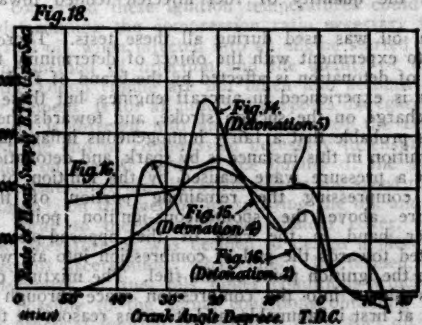


diagram. In Fig. 14 the detonation figure is 5 on the arbitrary scale adopted. In Fig. 15 the cams give a slightly greater period of opening. Fig. 16 is a combustion diagram with long-period cams in use. In this case the fuel consumption is greater, and the detonation less, than with the short-period cams.

From the combustion diagrams shown in Figs. 14 to 16, the pressures in excess of the pressures given by the no-fuel compression-expansion diagrams were plotted on an angle base, and from these curves the rates

of change of pressure were obtained. Fig. 17 shows the rate of pressure change curves. It will be seen that heavy detonation is accompanied by a high rate of change of pressure occurring just after the dead center.

From the combustion diagrams temperature-entropy diagrams were also constructed for comparative purposes, with the assumptions that the temperature of the air charge at the beginning of compression was 200 degrees F. and that the weight of the working fluid was constant throughout the cycle. From the temperature-entropy diagrams the heat supplied and the temperatures throughout the cycle were plotted for each case, and from the heat supply curves the rate of the supply of heat was obtained. Fig. 18 shows the curves of rate of heat supply. It will be seen that in the case of the combustion diagram shown in Fig. 17, where the detonation was heavy, the rate of the supply of heat was greater than in the case of the low detonation diagram, Fig. 16.

From the various diagrams constructed the following points were clearly indicated:—

1. An increase in the rate of the supply of heat, *i.e.*, in the rate of combustion of the fuel, resulted in heavier detonation.
2. In the case of low detonation diagrams heat was being supplied later in the expansion stroke than in the case of the high-detonation diagrams.
3. The maximum and mean temperatures of the cycle were greater in the low detonation diagrams than in the high-detonation diagrams.
4. The thermodynamic efficiency was higher in the case of the high-detonation diagrams than in the case of low-detonation diagrams (this agrees with the fuel consumptions recorded).

From the experiments which were carried out it was clear that detonation was a function of the speed of injection with the particular oil fuel used, but when working at mean indicated pressures just above 130 pounds per square inch an increase in the quantity of fuel oil injected, other conditions remaining the same, tended to decrease detonation—although in this instance the exhaust did not remain colorless. Also, a reduction in the quantity of fuel injected tended towards decreased detonation.

Texas fuel oil was used during all these tests. There has been no opportunity to experiment with the object of determining to what extent the intensity of detonation is affected by the brand of fuel oil used.

Detonation is experienced in aircraft engines, but these engines draw in the fuel charge on the suction stroke, and towards the end of compression it is probable that a fairly homogeneous inflammable mixture is obtained. Ignition in this instance is by spark, and detonation is generally attributed to a pressure wave, caused by the ignition of a portion of the charge compressing the remaining portion of the charge to a temperature above the spontaneous-ignition point of the fuel. On the other hand in the case of the opposed-piston engine the fuel is injected towards the end of compression into air whose temperature is above the ignition point of the fuel. The mixture of fuel oil and air which is blown into the compression space through the two fuel valves is not at first inflammable, and it seems reasonable to suppose that ignition does not occur until the speed of the incoming charge is reduced, its temperature raised above the ignition point of the fuel oil and the distribution such that an inflammable mixture is obtained. When heavy detonation occurs, therefore, it is probable that there is a combination of circumstances favorable to rapid combustion, *i.e.*, by the time the incoming charge of fuel and blast air has had its temperature raised above the ignition point of the fuel, either by admixture with highly heated air in the cylinder, by radiation from the small portion of fuel vapor

which commences to burn on the fringes of the jets or by the formation of a pressure wave (as in the case of the aircraft engine), a large proportion of the fuel has been distributed generally in the compression space and an inflammable mixture obtained, with the result that combustion is extremely rapid.

It has been mentioned that when working at high mean pressures an increase in the quantity of fuel injected, other conditions remaining the same, tended to decrease detonation. In this instance it is probable that the additional fuel injected resulted in the formation of pockets too rich in fuel, and that combustion was consequently delayed. Although many experiments were carried out, it has not so far been possible to run at high powers with Texas fuel oil at comparatively low fuel consumptions and at the same time avoid the detonation effect entirely. Generally speaking, a reduction in the intensity of detonation resulted in increased fuel consumptions. It is probable, however, that better results may be obtained by further modifications to the fuel valves and cams. Detonation is not very noticeable in the laboratory 20-inch by 20-inch air injection, four-stroke cycle, single-cylinder experimental engine when using Texas fuel oil and running at high mean pressures at about 300 revolutions per minute. It would appear, therefore, that the type of engine has some influence on the intensity of detonation. As, however, the maximum pressures were approximately the same in the four-stroke engine and in the two-stroke opposed-piston engine, it would seem to point to temperature as being one of the factors on which detonation depends, and not pressure.

So far as is known the spontaneous ignition temperatures of fuel oils have been determined hitherto, in either air or oxygen, at atmospheric pressure by allowing drops of the fuel oil to be tested to fall on to a heated porcelain or into a platinum crucible, and noting the temperature at which the oil drops burst into flame. Although the results thus obtained are of value, and no doubt can be regarded as the true ignition temperatures of the apparatus used, it was felt that so far as Diesel engines were concerned these ignition temperatures could not be used directly for determining the compression ratio necessary to ensure the ignition of a fuel oil, when injected into an engine started from the cold condition. The method of injection and the quality of the fuel spray obtaining in the Diesel engine are not taken into account in these tests and the "time element" is for all practical purposes disregarded altogether.

When fuel oil is injected into an oil engine of the Diesel type the temperature at the end of compression necessary to ensure spontaneous ignition depends on the oil used, on the quality of the spray (*i.e.*, the degree of pulverization of the oil, which affects the speed of vaporization preceding ignition), on the method of spraying the oil, and on the proximity of the spray to the water-cooled cylinder cover. Having these points in view, therefore, it was decided to design a piece of apparatus with the object of ascertaining, what might be termed, the practical ignition temperatures of fuel oils under conditions which approximate as closely as possible to the actual conditions obtaining in an engine. Knowing the practical ignition temperature of a fuel oil it was hoped that the lowest compression ratio necessary to ensure the ignition of the fuel in an engine, when starting from the cold condition, would be readily obtained from a temperature-compression curve.

A section through the cylnder and a diagrammatic arrangement of the apparatus is shown in Fig. 19. It will be seen that the apparatus consists of a steel cylinder A fitted at one end with a water-cooled cover B in which the fuel valve is situated. This cover is also fitted with

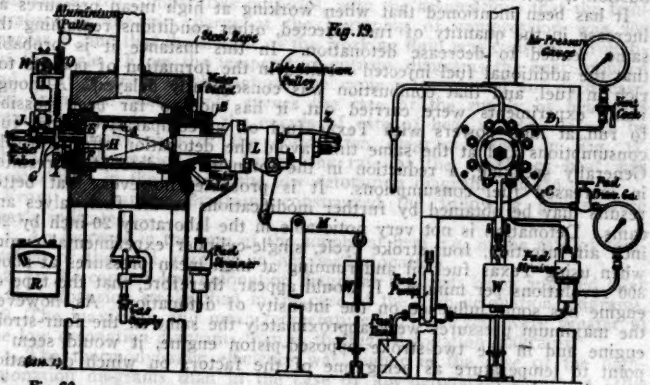


Fig. 20.

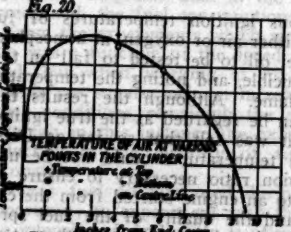


Fig. 22.

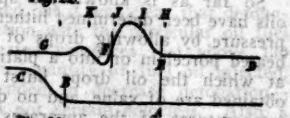


Fig. 23.

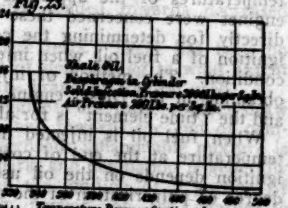
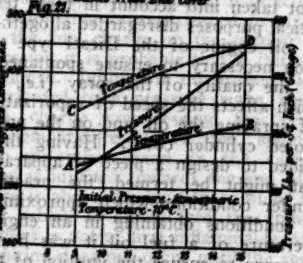


Fig. 21.



compressed air and gauge connections C and D. At the other end of the cylinder a steel plate cover E not water-cooled, is provided, which is held in position by the piston nut F and set screws. The cover E is fitted with a relief valve and indicator connection G and a heat-insulated sheath H which carries the thermocouple J. Provision was originally made for a fan to circulate the air in the cylinder and thus obtain a more uniform temperature before the injection of the fuel oil, but so far it has not been found of any great value.

The steel cylinder, which is 5 inches internal diameter and 10½ inches long, is fitted with a Brayshaw gas furnace, so that the compressed air contained in the cylinder can be brought to any desired temperature. When the cylinder is heated the temperature of the shell is slightly higher than the temperature of the air it contains, and in this respect, therefore, it differs from the water-cooled cylinder liner of an engine. The diameter was, however, fixed at 5 inches so as to ensure that with the fuel sprayer used the spray would not strike the heated shell.

It was intended that both the solid and the air-injection systems should be tested and the fuel valve L was designed with this purpose in view. The fuel valve was, however, at first arranged for the solid-injection system and, as it was desirable that the quantity of fuel oil injected should only be sufficient to ensure an appreciable rise in pressure after combustion, the sprayer was provided with one hole 0.013 inch in diameter.

The fuel valve is operated by means of a falling weight W, released by a lever (not shown in the figure), which, after falling through a distance of about 1 foot, strikes the washer Y secured to a rod attached to the valve lever M.

The fuel system is charged to the pressure required by means of a small pump placed near the apparatus.

To ascertain the distribution of the fuel oil with the 0.013 hole sprayer a wooden plug, covered with white paper, was placed in the cylinder so that its surface was at right angles to the axis of the cylinder. The cylinder was then filled with compressed air to a pressure of 400 pounds per square inch at atmospheric temperature, and when a charge of fuel oil was injected the distribution was found approximately from the diameter of the stain on the paper. By placing the plug in various positions, and injecting a charge of oil for each position, the distribution throughout the length of the cylinder was readily ascertained. It was found that when the plug was placed hard up against the cover E the paper was not completely covered with fuel oil, which indicated that little or no oil was striking the sides of the cylinder. In order that a false reading should not be obtained, however, by a portion of the spray striking the cylinder a sheet iron cup surrounded by an air space was fitted at a later date, as shown dotted in Fig. 19. The base of the cup was placed at the point of maximum temperature in the cylinder, *i.e.*, about 3 inches from the end, and the thermocouple sheath passed through the base of the cup. It was found that the fitting of the cup made no appreciable difference in the results obtained, but it was thought desirable to retain it.

The cylinder is charged with compressed air from the laboratory air main and heated up to the required temperature. The leakages of air from the fittings attached to the cylinder are very slight—the principal leak being past the indicator piston. This is overcome by charging the cylinder to a slightly higher pressure than that required for the experiment and injecting the oil as soon as the pressure has fallen to the required figure. As a matter of fact, it has been found that there is little, if any, difference in the results obtained whether the pressure of

air in the cylinder before combustion is 400 pounds or 200 pounds per square inch. Moreover, with the same sprayer it is found that at a given temperature the effect on the "time lag" is not very marked when the fuel pressure is varied between 2,000 pounds and 4,000 pounds per square inch. When the charge of fuel oil is injected into the cylinder and ignition has taken place, which is indicated by a rise in pressure and also by a slight rise in the temperature of the air, the cylinder is cleared out by relieving the pressure through the vent cock and blowing through the compressed air. In addition to the ordinary indicator for recording pressure rise a separate pencil attachment O is fitted, which is constrained to move vertically. This attachment is actuated by means of a steel tape 0.006 inch in thickness, which is connected to one arm of the bell-crank lever Z (Fig. 19), the other arm of the lever pressing against the end of the fuel-valve spindle. The movement of the fuel-valve spindle is therefore transmitted to the pencil and a valve-lift diagram is drawn on the paper carried by the continuous recorder N. A spring pulls the pencil attachment down when the fuel valve is closing, but it is only the instant of the opening of the valve which it is desired to avoid. The valve-lift pencil O is placed vertically above the ordinary indicator pencil, and valve lift and pressure diagrams are recorded on the same card. The indicator is placed about 8 inches from the cylinder cover to protect the spring from the heat of the furnace. The piston and cylinder of the indicator, as in the case of the indicators used with the experimental engines, are thoroughly cleaned after each test. It was found that the speed of the paper of the ordinary rotary recorder was insufficient, and a new indicator was obtained, giving a paper speed of 20 inches per second. Several timing devices were tried, but that now in use consists of a trembler coil, to which a pencil is attached, operated by a battery through a commutator driven by a constant-speed clock mechanism.

The temperature of the air in the cylinder is measured by a base-metal thermocouple J and an indicator R. The hot junction of the couple was made by brazing together the ends of "nichrome" and "ferry" metal wires. The cold-junction ends of the wires were fitted into glass tubes closed at one end and placed in water in a vacuum flask. The hot junction is placed in contact with the end of the sheath H, which is insulated from the cover with asbestos washers, etc. It takes an appreciable time to adjust the temperature of the air to that required for the experiment, and it is considered that the temperature of the end of the sheath, which is made as thin as possible, can, for all practical purposes, be taken as the temperature of the air in its vicinity.

The following experiment was carried out in order to ascertain the variation of the temperature of the air in the cylinder. The end cover E (Fig. 19), which corresponds to the piston of an oil engine, was removed and replaced by a thick sheet of asbestos millboard so that the position of the thermocouple could be readily varied. The air in the cylinder at atmospheric pressure was then heated up until a constant maximum temperature was reached. The thermocouple was then placed at various points on the center line of the cylinder, and also in contact with the sides of the cylinder. Three sets of readings were taken and the mean temperatures obtained for each position of the thermocouple, for a maximum temperature of about 350 degrees C., are shown in Fig. 20. From the curve showing the temperature gradient on the center line of the cylinder it will be seen that the maximum temperature is at a point about 3 inches from the air-cooled cover, and for a distance of 1 inch on either side of this point the temperature variation is not more than 5 degrees C.

The greatest variation of temperature of the cylinder wall, as recorded by the couple placed close up to the wall, occurred at the top and bottom of the cylinder—the top being somewhat hotter than the air on the center line. At a distance of 4 inches from the air-cooled cover the temperature of the air in contact with the cylinder top was about 10 degrees C. above that of the air on the center line. This was the maximum temperature recorded. By making each ignition experiment on a very slowly falling temperature it is probable that such differences would be lessened, if not entirely eliminated. During the experiments the couple is placed at a distance of 3 inches from the end cover E, Fig. 19, and it is considered that the results obtained are accurate within ± 5 degrees C.

In a Diesel engine the temperature at the end of compression when starting from cold must be sufficient to ignite spontaneously the injected fuel oil in the time available. The initial temperature of the working fluid at the commencement of compression will be appreciably higher when the engine has been firing than when it starts from the cold condition. The cold condition is consequently the determining factor in regard to the compression ratio required for a given fuel oil, and in these experiments, therefore, the starting condition only has been considered. Assuming the equation $p v^n = \text{constant}$ between the initial and final compression pressures, the average value of n obtained from a number of compression cards taken from a four-stroke engine was 1.27, with compression ratios of 11 : 1 and 12.1 : 1. This engine was started with compressed air from the cold condition, and when the speed had reached that usually necessary to ensure the ignition of the injected fuel the starting air was shut off and the compression card taken.

Curves showing the temperature and pressure at the end of compression for various compression ratios are shown in Fig. 21. These curves are based on the following assumptions:—

(1) That the temperature of the air in the cylinder at the commencement of compression, when the engine is cold, is 10 degrees C. (50 degrees F.).

(2) That the volumetric efficiency when the engine is running on starting air is 100 per cent.

(3) That the index n of the compression curve is 1.27.

A number of combustion diagrams were taken when starting the four-stroke experimental engine referred to above from the cold condition, using shale-oil fuel with a compression ratio of 11 : 1, and it was found that to ensure an easy start the average time which elapsed from the point of opening of the fuel valve to the point at which the pressure began to rise above the "no-fuel compression-expansion" curve had to be of the order of 0.04 seconds. In the experiments which were carried out in the ignition-temperature apparatus, therefore, it was assumed that the ignition of the fuel-oil, as indicated by the rise in pressure, had to take place in 0.04 seconds from the instant the fuel valve commenced to lift.

In the preliminary experiments with the ignition apparatus a temperature was selected for an experiment, and the cylinder was filled with air at a pressure slightly higher than the pressure corresponding to the temperature obtained from Fig. 21 (full lines). The air was then heated up until the reading of the temperature indicator was steady at the desired temperature. As soon as the pressure of the air in the cylinder had fallen to the pressure corresponding to the temperature the indicator drum was set in motion and the weight, W, Fig. 19, allowed to fall, thus injecting a small quantity of fuel-oil into the cylinder. The pressure in the fuel-oil system had, of course, to be previously pumped up to that required for the experiment. The time which elapsed between the instant

of opening the fuel valve and the instant the rise of pressure occurred, due to the combustion of the oil, was measured. If this differed from the time interval allowed further experiments were made at different temperatures and pressures until the correct time interval was recorded. By this means what has been termed the practical ignition temperature was obtained for the particular oil with the assumptions made.

Temperature of air before ignition	349 degrees C.
Rise of air temperature after ignition	5 degrees C.
Air pressure before ignition	200 pounds to square inch.
Rise of pressure after ignition	170 pounds to square inch.
Fuel oil pressure	3,000 pounds to square inch.

A portion of a diagram obtained from the apparatus, when using solid injection, is shown in Fig. 22. The oil used was shale fuel-oil having a specific gravity of 0.86 at 60 degrees F. The curve A B C shows the pressure in the cylinder. The pressure at A is 200 pounds per square inch, and the pressure at C, after the ignition of the oil, is 370 pounds per square inch—a rise of 170 pounds per square inch with the quantity of fuel-oil injected. The curve D E F G shows the fuel-valve lift, magnified in the original diagram about six times. The fuel valve opened at E and closed at F. The points H I J K, etc., were marked on the diagram by the timing device, and the average time interval is 0.019 seconds, *i. e.*, the indicator paper traveled 1 inch in 0.052 seconds. The pressure in the fuel system was kept constant at 3,000 pounds per square inch.

It will be seen from this diagram that the indicator begins to show a rise of pressure in the cylinder at the point B, and the distance A B corresponds to a time interval of 0.068 seconds, *i. e.*, the time which elapsed between the commencement of opening of the fuel valve and the indications of rise of pressure in the cylinder was 0.068 seconds. The temperature of the air in the cylinder when the oil was injected was 349 degrees C.

As the temperature in the cylinder of the ignition apparatus is increased the "time lag," represented by A B in Fig. 22, decreases. Fig. 23 shows the "time lag" so far obtained with shale fuel at various temperatures. It will be seen that at about 350 degrees C. the "time lag" increases rapidly as the temperature falls. The lowest temperature at which there was any sign of ignition shown on the indicator card was 260 degrees C., the lag being 3.5 seconds.

With shale fuel-oil, when using solid injection, it has so far been found that a temperature of about 380 degrees C is required for a "time lag" of 0.04 seconds, but this temperature is much higher than the temperature at the end of compression of the experimental engine (Fig. 21) with a compression ratio of 11 : 1—and the engine will start readily from cold with shale fuel-oil using this compression ratio.

In a Diesel engine, owing to the cooling effect of the piston, cover, and liner, the temperature of the air near the walls at the end of compression is less than the temperature of the air in the center of the combustion space, *i.e.*, there is a hot zone in the center of the combustion space, and it is through this hot zone that the fuel is sprayed. The temperature of the end of compression from Fig. 21, which is derived from the formula $p v^{1.4} = constant$, is a measure of the mean temperature of the combustion space—but it is the maximum temperature of the combustion space, *i. e.*, the temperature of the hot zone, which causes the fuel-oil spray to ignite. It would appear, therefore, that the temperature obtained from the ignition apparatus is more a measure of the maximum temperature of the hot zone in the combustion space of the engine, and consequently the temperature-compression ratio curve A B in Fig. 21 cannot be used to

ascertain the compression ratio to be employed to ensure an easy start from cold with a given fuel-oil. It is probable that the curve C D, shown dotted in Fig. 21, would be more nearly correct for this purpose, but this question is being further investigated. The maximum temperature of the hot zone in an engine, is of course, dependent on the shape of the combustion chamber.

Although a large number of tests have been made with shale fuel-oil, to ensure the satisfactory working of the apparatus, there has been no time to carry out many tests with oils of other brands. So far as the experiments have gone with shale-oil fuel it has been found that for the same quantity of fuel-oil injected an increase in the initial temperature of the air in the cylinder resulted up to a certain point in an increase in the maximum pressure rise and in a reduction in the "time lag." Above this point an increase in the initial temperature reduced the "time lag," but the maximum pressure rise due to the combustion of the fuel-oil began to decrease. This would appear to indicate that combustion was delayed at the higher temperatures, and it confirms the results obtained with the four-stroke engine during the tests with the hot plates, referred to on page 151. A few experiments were made with shale fuel-oil when using air injection, and it was found that it was necessary to have a temperature of about 40 degrees C. higher than the temperature with solid injection, for the same "time lag."

As previously mentioned some marine Diesel air compressors have been a continued source of trouble. Although there are many points to be considered the author would like to mention two, which, in his opinion, have been largely responsible for the difficulties which have been encountered in certain cases.

In nearly all cases a Diesel compressor is provided with sufficient capacity to charge the starting-air bottles as well as to provide the requisite quantity of blast air, and it frequently happens that the compressor is not designed from the point of view of the work it will be expected to perform during the greater part of its life. It has also been noticed—and a case occurred recently—that when a firm specializing in compressors is asked to supply a machine for coupling to a Diesel engine the only items generally specified are the maximum output required in free air per minute (which is fixed so as to ensure a good margin), the revolutions of the compressor and the maximum working pressure. No mention is made of the volume of air which will ordinarily be necessary for blast purposes. In consequence of this the compressor is designed for the full output, whereas most of its time it will be running at a reduced output for supplying blast air only.

As an example, let it be assumed that a three-stage compressor is designed so that the theoretical final temperature is the same in each stage. If the clearance volumes in the first, second, and third stages are taken as 4 per cent, 4 per cent and 5 per cent, respectively, and it is assumed that the air-suction temperature is 70 degrees F., and that it is cooled to 70 degrees F. between the stages, the temperature at the end of each stage, obtained from theoretical diagrams, would be about 280 degrees F. for a final pressure of 900 pounds per square inch. If the compressor supplies blast air only, which is assumed to be seven-tenths of the full output, the maximum theoretical temperature will rise to 336 degrees F. owing to the increased ratio of compression. When supplying blast air only under working conditions, with tight valves, temperatures higher than 336 degrees F would be reached on account of the increase in pressure necessary to cause the flow of air and to overcome the frictional resistance of valves and passages, dependent on the capacities of the intercoolers and

on the arrangement of the stages. Further, it is also probable that after the first stage the initial temperatures of the second and third stages would be higher than 70 degrees F. There is no doubt that in certain circumstances very high final temperatures are obtained and small explosions in pipes, etc., which have affected the good working of the valves, have not been exceptional.

From the point of view of maximum air temperatures, therefore, it is considered preferable to fix the staging so as to get the minimum stage temperatures when the compressor is supplying blast air only. As an alternative, should the general arrangement of the machinery permit of this, a four-stage compressor could be fitted. In the alternative scheme the first stage would be of sufficient capacity to give the full output required for charging the starting-air bottles and providing the blast air, and the second stage would be of sufficient capacity for providing blast air only, with a small margin. When charging starting-bottles all stages would be in use and there would be four stages of compression, but when providing blast air only the first stage would be cut out entirely and the second, third, and fourth stages would operate as a three-stage compressor —the second stage then taking its suction from the atmosphere. The dimensions of the third and fourth stages would be fixed so as to give the minimum final temperatures in the second, third, and fourth stages when supplying blast air only. It is not necessary to go into the question of the arrangement of suction pipes with this proposal but no difficulties are anticipated.

With the four-stage proposal, when supplying blast air only, the second stage virtually becomes the first stage of a three-stage compressor, and it is pumping at practically its full output. The difference in volume between the first and second stages would, of course, depend on the extra output required for charging starting-air bottles. In the case of a compressor attached to a two-stroke engine, when the scavenging pressure is from 3 pounds per square inch to 4 pounds per square inch, a three-stage compressor could be fitted, and arrangements made so that when charging starting-air bottles the first stage would take its suction from the scavenge air supply instead of the atmosphere—which is equivalent to the four-stage proposal outlined.

An excess of lubricating oil in the compressor cylinders, combined with the comparatively high temperatures referred to above, is another frequent source of trouble. In a number of designs, particularly in the case of high-speed engines with enclosed crank-cases, the first-stage cylinder of the compressor is open to the crank-case, and the piston is of the ordinary trunk type. Guards are frequently fitted over the compressor crank, but they do not prevent the lubricating oil, especially from the first-stage piston gudgeon pin, finding its way on to the first-stage cylinder liner.

In the case of enclosed engines, there is always an oil mist in the crank-case. It has been mentioned that the period during which the compressor is used for the dual duty of charging starting bottles and supplying blast air is a comparatively small fraction of the time the compressor is working. For the greater part of its time the compressor is, therefore, running at reduced output, and, the throttle in the first-stage suction pipe being partly closed to effect this, the resulting partial vacuum in the first-stage cylinder facilitates the passage of lubricating oil and oil mist past the piston. In cases where the first-stage cylinder is open to the crank-case it is considered that better results would be obtained by—

1. Removing the throttle from the first-stage suction.
2. Fitting a drain bottle of suitable capacity between the first-stage cooler and second-stage suction.

3. Fitting an overflow valve to the drain bottle.

4. Fitting to the first-stage piston a suitable guard with a drain so that the lubricating oil from the gudgeon pin falls clear of the compressor cylinder.

The overflow valve on the drain bottle would be the means of controlling the supply of air. The first stage would always work at its full capacity, and there should be less lubricating oil drawn into the cylinder. It is probable, however, that a small quantity of oil would still find its way in—in fact, a certain amount would be necessary for ordinary lubrication—but a proportion of this oil would be discharged through the overflow valve on the drain bottle, which would always be open except when pumping up the starting bottles.*

It might be mentioned that difficulties were experienced with the compressor of a certain high-speed engine, and it was found that the quantity of lubricating oil which found its way into the cylinders was excessive. More effective guards, etc., were fitted which reduced considerably the quantity of lubricating oil which passed from the crank-case into the compressor cylinders. Since the alterations were made the compressor has given no trouble.

In the case of high-speed reciprocating steam engines fitted with forced lubrication great care is taken, although for another reason, to prevent lubricating oil finding its way into the cylinders, and it is a question well worth considering whether compressors for Diesel engines should not be treated in a similar manner. This would slightly increase the first cost of the Diesel engine, but the cost of upkeep, so far as the compressor portion is concerned, should be less, and the engine should be more reliable.

It is considered, therefore, that by giving closer attention to the staging of the compressor, and also designing and arranging it with the view of reducing the chances of lubricating oil passing into the cylinders in excessive quantities, the troubles with compressors should be considerably reduced, if not entirely eliminated.

The air compressor question has frequently been considered at the Admiralty Engineering Laboratory, and an arrangement has been devised following on the lines of Trinkler's proposal. Should such a system be found in every way satisfactory it would result in an appreciable reduction in the number of vital parts in a compressor.

The experimental work which has been outlined in this paper is concerned principally with the high-speed engine, but the injection of fuel-oil is equally important in the case of the slow-speed engine, and several of the questions dealt with are common to both types. In some cases, particularly in the case of the ignition experiments, the results given are not as complete as one would wish, as the experiments are proceeding, but it is hoped that what has been given will prove of interest to the members of the Institution, and that the reasons advanced for certain occurrences will stimulate discussion.

In conclusion, the author desires to express his thangs to the Admiralty for permission to read this paper, and to the staff of the Laboratory, especially to Mr. Newman, Chief Designer, and Mr. Aitken, Chief Testing Engineer, for their valuable assistance in connection with the experimental work. The author also wishes to acknowledge the valuable assistance rendered by Dr. O. F. Hudson in connection with the ignition experiments.—"Engineering," Dec. 3 and 10, 1920.

* In the air compressor constructed by Messrs. Reavell the supply of air is controlled by means of the drain valve between the first and second stages.

MECHANICAL REDUCTION GEARING.

The rapid and almost universal adoption of the mechanical reduction gearing in conjunction with high-speed turbines for the propulsion of all classes of ships is certainly one of the most remarkable and most complete revolutions in the whole history of the development of marine engineering. Within a space of no more than 10 years, from the position of being almost untried, the mechanical reduction gear has come to be most extensively employed. Such a result, of course, could never have been achieved unless the idea of employing gear wheels between the turbine and the propeller had met a long felt want, and the fact that it enabled the efficiency of both to be greatly increased as compared with when they were directly coupled together added no doubt to the attractiveness of the idea. Moreover, the rapidity with which engineering science solved the problem of manufacturing the wheels and pinions even when the highest powers were being transmitted is to a large extent responsible for the success which has attended the employment of mechanical gearing.

There is, however, a distinct possibility that this complete and rapid success may not prove ultimately to the benefit of the idea of using mechanical gearing. There is already noticeable a tendency for the type of gearing employed to become standardized, and as a result, improvements either in design or in methods of production are not being made. The reason for this is not far to seek. When it is remembered that the efficiency of transmission obtained with gearing as produced today is somewhere in the region of 97 per cent, it seems difficult to imagine that any improvements can be made; but it will be at once admitted that it is not only in the direction of increase of efficiency that we are to seek improvements; there are such problems as the space occupied by the gearing, its weight, its lubrication, its position relatively to the turbine, etc., all of which are worthy of the closest study with a view to improvements being effected. Again, the question of cost is of prime importance, and there is always plenty of scope for the introduction of improvements which will materially reduce the cost of production of the complete set of gearing without in any way sacrificing efficiency or durability.

At the present time the double-helical form of gearing is almost invariably employed in which the pinion on the turbine shaft drives either through one set of wheels or two on to the propeller shaft, thus giving the required ratio of speed reduction. It is generally admitted that with high-speed turbines running at somewhere in the region of 4,000 revolutions per minute a double reduction gear must be employed if an economical propeller speed is to be obtained. The double reduction gearing is, however, obviously heavier, more costly, is not so efficient and occupies more space than does the single reduction gearing, and for this reason many makers are inclined, even at the sacrifice of a certain amount of turbine efficiency, to adopt single reduction gearing. It would prove of the utmost value if some comparative data could be obtained giving the total efficiency of turbine, gear and propeller with vessels of the same class but in which some have single reduction and others double reduction gearing. Again, a considerable amount of diversity of opinion exists as to the best arrangement of the turbine and propeller shafts relatively to one another. It is now widely recognized that the center lines of the shafts need not be in the same horizontal plane, and the pinions can be placed anywhere around the periphery of the wheel according to the general lay-out of the engine room.

One of the great disadvantages of the single reduction gear lies in the fact that it involves the employment of a gear wheel of large diameter,

thus tending to unduly raise the positions of the propeller shaft and the turbine. While, however, this disadvantage will be admitted, it may be made a source of gain or economy if the turbine be mounted sufficiently high, so as to allow the pinion to run on the top of the gear wheel, and sufficient space is provided to enable the condenser to be placed immediately below the turbine, being mounted on the condenser so that the turbine exhausts direct into it. It will be seen that the long length of exhaust pipe usually found in a ship's engine room is now eliminated, there being a considerable increase in the vacuum with corresponding gain of economy in operating the turbine. There is also the gain of space in the engine room. This method of mounting the turbine immediately over the condenser is generally adopted in electricity generating stations on land where the economy of operating the turbines reaches a very high figure. The conditions on board ship are different, of course, but it should be possible to introduce this principle with considerable advantage.

But while the efficiency of the mechanical reduction gearing, as at present made, is far and away superior to other forms, it is just as well to remember that the electric and the hydraulic do exist, and that these have one or two features which strongly recommend them to marine engineers. Thus, with both the hydraulic and the electric systems, the simplicity with which the propeller can be reversed without affecting the direction of the main turbine is a feature which is of great value, particularly in the case of war vessels. Again, the flexibility of these systems, that is the range of propeller speeds which can be obtained without calling for a similar variation in the speed of the turbine, is also a feature which recommends these systems again particularly to the naval engineer. Thus, while the mechanical reduction gearing has proved itself so far to be far superior to all other forms, it is not impossible that a close study of these other systems might reveal methods whereby the required end of economical turbine and propeller speed could be obtained in a far less costly way than at present. Actual efficiency of transmission is not all that is required, for if a system of gearing could be devised which in cost, weight, space occupied, reversibility and flexibility proved itself to be far superior to the system now so widely employed, we should be prepared to sacrifice a certain amount of efficiency of transmission in order to obtain these and other desirable ends.—"Shipbuilding and Shipping Record," Nov. 11, 1920.

THE MEASUREMENT OF PROPELLER THRUST.

Among the appliances which are now found useful for the proper equipment of ships, not so much for the purpose of increasing their earning powers as to give guidance along scientific lines, that which records the actual thrust produced by the screws must soon come into more general use. In an interesting paper read to the Institution of Naval Architects at Liverpool in July last, Mr. Frodsham Holt gave an analysis of the results of the destroyer *Mackay*, in which vessel records of the actual thrust had been taken. When by means of torsionmeters the actual shaft horsepower delivered can be measured and by means of thrust recording apparatus the actual thrust is obtained a close estimate can be made of the actual efficiency of the screw.

To Mr. Hamilton Gibson, of the firm of Cammell, Laird & Co., belongs much of the credit for making this measurement possible. In a paper read to the Institution of Naval Architects in 1907, on the subject of Torsionmeters, he remarked on the importance of eliminating unknown and vari-

able factors, such as engine friction, in arriving at the actual shaft horsepower delivered to the propeller, and at the same time suggested that the investigation should be carried a step further by the use of simple apparatus that would record the actual thrust along the line of shafting, and enable a direct comparison to be instituted between the power transmitted to and the power produced by the propeller. An opportunity presented itself for attempting this measurement, and in the first instance spring balances were used for taking the whole propeller thrust. The wide oscillations of the balance pointers, however, made it impossible to obtain reliable readings. Following this attempt a hydraulic method was tried with complete success. Sir Archibald Denny has stated that his firm has also made accurate thrust measurements, but no records of such, so far as the writer is aware are in public print.

Having obtained this thrust, the problem arises as to how to make use of the information. Mr. Holt's paper is an answer to this query. To begin with, the effective horsepower of the vessel should be known. This effective horsepower is the power actually necessary to overcome the resistance of the ship, *i.e.*, a particular resistance in tons requires to be overcome if a certain velocity is to be obtained. On the average it is found that it is necessary to produce double the horsepower that this consideration of speed and resistance suggests, and this is caused mainly by propeller losses. The actual resistance of a vessel cannot be measured, so recourse is made to experiments on a model of the ship, and from the recorded resistance of this model that of the full-sized ship is calculated. In the case of a twin or quadruple screw ship, either some allowance is made for the resistance of the appendages or the model is run with bossings, bilge keels and rudder fitted thereon.

It is known that the thrust required must be greater than the resistance of the ship, because of the augment to the resistance caused by the suction action of the screws on the water in front of them. The difference between the recorded thrust and the actual resistance is the amount of this augment to resistance, which is otherwise known as the thrust deduction. This value is one of the factors in what is known as the hull efficiency. The latter term is applied to the gross result obtained through associating together the advantage gained in propulsion through the presence of the wake and the apparent loss due to this increase in resistance just referred to, namely, the thrust deduction. In order, therefore, that the hull efficiency value can be arrived at it is necessary to know the speed of advance of the water in which the screws are working, viz., the wake speed. All a propeller is called upon to do is to produce a thrust equal to the augmented resistance. The thrust horsepower is proportional to this thrust multiplied by the speed of advance of the propeller in relation to this wake. It follows then that the greater the speed of the wake is, the less becomes the thrust horsepower necessary to maintain the speed of the ship. Thus the thrust horsepower may be much less than the effective horsepower. In such a case the hull efficiency value is greater than unity. This gain through large wake values may be more apparent than real, but this need not be discussed here. In order then to arrive at the thrust horsepower it is necessary to know the wake speed. This cannot be done by actual measurement, but is estimated from the performance of model screws of the same pitch and area ratios as that of the full-size screws.

It is found that the model screws show a higher efficiency than that given by the full-sized propellers. In the case of the *Mackay's* analysis what is called the "rotative" efficiency was introduced to take account of this difference in performance between the experimental and actual screws. From considerations of the effective horsepower at a particular speed and

the shaft horsepower actually necessary to obtain that speed the propulsive efficiency is known. Neglecting the small correction for loss in power between the point where the torsion is actually measured and the screw, this propulsive efficiency should equal that of the propeller multiplied by the hull efficiency value. It is found, especially at low speeds, that the actual propeller efficiency is very small, very much less than that indicated from the performance of model screws. It is difficult to find a reason for the full-sized propeller being less efficient than the corresponding model screw, and although this "rotative" effieiency value takes account of it, it gives no guidance as to its cause.

Mr. Hamilton Gibson suggested that the design of propellers would soon be made on the basis of production of stated thrust, and with this view the writer is inclined to agree, for the thrust horsepower—the basic consideration in Froude's propeller curves—is a very elusive value, whilst the actual thrust to be developed can be very closely estimated.

Propeller design is a really difficult task, although screws approximating to those of greatest efficiency can be easily enough arrived at, but the dimensions of the screws to give the highest efficiency to fulfil stated conditions are difficult to find.—"Shipbuilding and Shipping Record," Oct. 7, 1920.

RECENT ADVANCE IN OIL BURNING.*

By Ernest H. Peabody, Esq., Member.

In the few years which have passed since a paper on the subject of oil fuel was presented before this Society, the great World War has been fought and won by the powers opposed to barbarism—won so recently, in fact, that the question "who will win the peace" is still debatable.

During this war the use of oil for the generation of steam has assumed an unprecedented importance, and I do not think it wide of the mark to say that the next great war will be won by the nation most plentifully supplied with oil fuel; at least it cannot be won by any nation whose supply of oil will not successfully meet a stupendous demand. It must be hoped that our own government will immediately and adequately recognize this condition.

But it is not with the supply of oil that I intend to deal, except to say that I am reliably informed by those in a position to know that the American driller, the American tanker, and the American dollar will assure this country of the oil it needs, if they are properly backed by the American people.

In speaking of the next war I may be charged with being a reactionary. I hopefully share the view that some day international warfare may be controlled to a degree proportionate to the success with which private warfare has already been reduced in civilized nations, but the whole scheme of nature shows that to survive one must be prepared to fight, and preparedness today largely signifies the ability and readiness to organize and use natural forces to the limit of skill and ingenuity.

Regret it as we may, the competitive necessity of warfare inspires more rapid scientific progress than times of peace. It is not strange, therefore, that we may look to the military arm of our government for the first signs of advance in the use of natural forces. In time of war they face the keenest competition of other nations, and in time of peace this com-

*Read at the twenty-eighth general meeting of the Society of Naval Architects and Marine Engineers, held in New York, November 11 and 12, 1920.

petition is in no way lessened even if it be potential in character. Nowhere is this leadership better illustrated than in the now universal use of the mechanical atomizer on oil-burning ships. First exploited in vessels of war, it has found a fertile field in the merchant marine.

The United States Navy is again pointing the way to future development in oil burning. In co-operation with leading mercantile interests who are manufacturing equipment for the Navy, they are experimenting not only in the use of heavy viscous oil but are demanding larger units and greater individual capacity both in boilers and in oil burners. I predict that this increase in size and rate of forcing will mark the trend of future development in the merchant marine. Today there has been no marked advance along these lines in the merchant service, notwithstanding the fact that the number of oil-burning ships has been enormously increased and our imagination has been stirred by the fitting up for oil fuel of some of the largest transatlantic liners.

I am indebted to the American Bureau of Shipping for permission to use the following estimate of the number of oil-burning vessels as compared to those burning coal. Under date of September 15 they state:—

"1. For the coal and oil-burning data of vessels now operating, the following has been compiled from various sources, none of which gives tonnage figures. It is to be assumed, however, that authentic tables, being used in comparison with those that are incomplete, will give a good knowledge of the information desired.

"2. United States Shipping Board vessels, which make up the bulk of American vessels, are daily changed from coal burners to oil.

"3. On August 1, 1920, there were documented in the United States the following number of steel and wooden vessels of 500 gross tons and up: 2,622 vessels of 10,535,922 gross tons. Of these the U. S. S. B. owned 1,662 vessels of 7,074,971 gross tons, thus leaving to private owners 960 vessels of 3,460,951 gross tons.

"4. No comprehensive data being available as to the fuel of these ships we will make a comparison with those facts available, as of January 1, 1920, pertinent to U. S. S. B. vessels only, *i. e.*:—

	Vessels.	*Per cent.*
Coal burners	841	56.75
Oil burners	635	42.5
Non-propelled	7	.75
	1,483	

"Applying this ratio to the present Shipping Board holdings of 1,662 vessels, we find:—

	Vessels.
Coal burners, approximately	945
Oil burners, approximately	705
Non-propelled, approximately	12
	1,662

"Of the 960 privately owned ships, the same method must be applied. The American Bureau of Shipping report of September 1, 1920, on steel vessels building in the United States to private account, having authentic figures relative to the fuel of these vessels, will be of value. It follows:—

	Vessels.	Gross tons.
Cargo, coal	20	38,171
Cargo, oil	101	400,455
Cargo, coal and oil	13	74,567
Cargo, non-propelled	109	82,350
	243	595,543

	Vessels.	Gross tons.
Tanker, coal		
Tanker, oil	121	830,964
Tanker, coal and oil		
Tanker, non-propelled	19	11,991
	140	842,995

Grand total, 383 vessels; 1,438,498 gross tons.

"6. Summarizing, we find:—

	Per cent.
Coal burners	5.2
Oil burners	58
Coal and oil burners	3.4
Non-propelled	33.4

"7. Applying these percentages to the 960 ships in question, we find:—Coal burners, approximately, 48 vessels; oil burners, approximately, 555 vessels; the balance being mainly non-propelled.

"8. It is interesting to note at this point that of the tonnage (steel) now building to private account, more than 58 per cent is made up of tankers.

"9. Recapitulation of the above data brings out the following:—

Operating:

	Vessels.	Per cent.
Coal burners, approximately	993	37.8
Oil burners, approximately	1,260	48
Balance (mainly non-propelled)	369	14.2
American vessels	2,622	

(Signed) "CROWLEY,
Statistician, ABS."

Thus, while very large vessels are using oil fuel successfully and from the estimate of the American Bureau of Shipping it is evident that nearly 56 per cent of the present American self-propelled merchant fleet consists of oil-burning vessels, it is probable that in not one of these ships is there a boiler containing more than 4,500 square feet of heating surface in the individual unit and no single oil burner capable of atomizing over 600 pounds of oil per hour, while the oil burned per hour per square foot of heating surface under forced-draft conditions will not equal one-half pound.

In other words, the present boiler and oil-burner practice in the merchant marine is following closely along the very conservative lines of previous custom, and the full value of the use of oil fuel has not yet been approached. Owners and designers have been content with the many great savings and advantages of oil fuel without perhaps realizing all its possibilities in the line of increased capacity without serious loss in economy.

There are many reasons for this. In the first place a large number of vessels now burning oil were originally designed for coal, and in others, which were designed to burn oil, the old proportions applying to the coal-burning ships have been adhered to, in many instances probably because it is feared that a return to coal fuel may some day become necessary. Furthermore, in the large majority of instances the vessels are fitted with Scotch boilers, and the above figures mark practically the limit to which such boilers may be forced with safety. The same statement will apply also to water-tube boilers until such time as more consideration is given to the type of condensers and the size and design of evaporators.

In marked contrast to the merchant marine, the Navy Department is making rapid advance of a most important character. It is now using water-tube boilers entirely in fighting ships, with improved condensers and evaporators. Boilers are being constructed containing more than 11,000 square feet of heating surface, while in the matter of size and capacity of oil burners, within the last few weeks successful tests have been run at the Fuel Oil Testing Plant at the Philadelphia Navy Yard in which more than one ton of oil was atomized per hour in the individual burners.

If then, as I believe, the work of the Navy is a forecast of what will eventually be the trend of development in the merchant marine, not only will considerable improvement be made in oil-burning equipment, but the Scotch boiler, with its enormous weight, poor circulation and unsuitability to high pressure, will become a thing of the past. This type of boiler has long been tottering on its last support of prejudice and the reactionary belief that it is impossible to keep salt water out of the feed system. We shall have, in its place, large, safe, light, economical water-tube boilers, installed in closed fire-rooms, burning oil and operating under forced draft with a few large-unit mechanical atomizers.

The remarkable economy of the properly designed small tube express-type water-tube boiler at high rates of forcing with oil fires was demonstrated not very long ago in some notable tests at the Fuel Oil Testing Plant. The tests were conducted under the direction of Lieut. Commanders A. M. Penn and W. R. Purnell, U. S. Navy, and their staff of trained assistants. The boiler was of the White-Forster design, built by The Babcock and Wilcox Company, and contained 7,565 square feet of heating surface and 753 square feet of superheating surface, the furnace volume being 751 cubic feet. Three different types of air registers were used:—the Bureau of Engineering forced-draft design (of which there were sixteen units installed), the Bureau of Engineering natural-forced-draft design (eleven units), and what was then known as the Peabody register manufactured by The Babcock and Wilcox Company, designed for either forced or natural draught and of which there were also eleven units. All three registers were tested successively with bureau burners. Final tests were then made with Peabody burners with Peabody air registers only.*

Through the courtesy of the Navy Department I am able to include in this paper the official results of the last two series of these tests, Table I, and I have taken the liberty of giving the plotted efficiency results of all four series in Plate I.

It will be noted that in the highest capacity test of June 10, 1919, there were eleven burners in operation, each atomizing 1,032 pounds of Navy Standard Oil (25.5 Baumé) per hour, giving a consumption of 1.5 pounds of oil per square foot of heating surface per hour. The evaporation of water per pound of oil from and at 212 degrees F. was 15.14 pounds,

*This equipment is now designated as The Babcock and Wilcox Air Register and the Babcock and Wilcox Mechanical Atomizer.

giving an evaporation of water per square foot of heating surface per hour of 22.73 pounds from and at 212 degrees with an efficiency of 76.15 per cent. The air pressure in the closed fire-room was 9.5 inches and the rate of combustion per cubic foot of furnace volume reached the very high figure of 15.12 pounds of oil per hour. It is believed that this test stands as a world's record for efficiency at high boiler and furnace capacity.

In the burning of oil there are many things more important than the mere rate of combustion as measured in pounds of oil per square foot of heating surface. The latter applies rather more to the boiler, and the ultimate capacity is a function of efficiency in heat absorption as well as the amount of oil burned. But it is interesting to note that tests have been made in England by Babcock & Wilcox, Ltd., in which the rate of combustion practically equalled 1¼ pounds of oil per square foot of heating surface per hour. I am indebted to The Babcock and Wilcox Company of New York for the following memorandum concerning these tests. Under date of September 9, 1920, my friend and former colleague, Capt. Walter M. McFarland, writes:

"The test of a Babcock and Wilcox boiler was made at Renfrew in the early part of 1917. This boiler was composed almost entirely of small tubes and was entirely experimental, as a first step in a series of studies.

"Six tests were made, in which the oil was burned at a rate of 1.2 pounds per square foot of heating surface per hour, or higher; one test was at the rate of 1.53 pounds with an air pressure of 2 inches, and the last was at the rate of 1.74 pounds with an air pressure of 2.08 inches. In the former case the baffling was of checkerwork on the top of the tubes, while in the latter there was no baffling at all." The oil burned per cubic foot of furnace volume at the highest rate was 11.64 pounds per hour.

In a paper before the Institution of Naval Architects presented by Mr. Harold Yarrow some years ago, reprinted in the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS for May, 1912, a test on a Yarrow boiler was reported in which the figures 1.94 were given as the rate of combustion in pounds of oil per square foot of heating surface per hour. It is to be noted, however, that in this test a damper in the uptake over one bank of tubes was closed and that the above high rate of combustion was reported on the assumption that only half of the heating surface was making steam. This assumption is, of course, unjustified, and it appears that the highest rate obtained in the Yarrow test was 1.24 per square foot of heating surface.

Impressed as I am with the importance of naval experiments, I have made an effort to secure some information as to what was being done in the development of oil burning by the British Admiralty. In this I have not been successful, but I am informed that a paper dealing with this subject is to be presented by Sir George G. Goodwin, Engineer in Chief of the British Admiralty, before the Institute of Naval Architects next February.

Returning to the work of the United States Fuel Oil Testing Plant now under the command of Lieut. Commander Purnell, it seemed at first sight that 1,032 pounds of oil per burner per hour was rather a notable achievement. It is interesting to state, therefore, that this record has already been materially outstripped. Early in the past summer The Babcock & Wilcox Company offered six air registers of their standard design, but of an enlarged type, with mechanical atomizers capable of atomizing 1,500 pounds of oil per unit per hour. These were tested by Commander Purnell under the same White-Forster boiler already described. The six burners successfully sprayed over 9,000 pounds of oil

per hour under conditions similar to those pertaining to the U. S. scout cruisers now under construction. The oil per square foot of heating surface was approximately 1.2 pounds, and the oil per cubic foot of furnace volume per hour about 12 pounds.

This record was exceeded in August in a test of the Normand boiler at the Fuel Oil Testing Plant. This boiler contained 4,500 square feet of heating surface and 487 cubic feet furnace volume and was fitted with three of the recently developed Peabody-Fisher wide-range mechanical burners and new Peabody air registers. Crude Mexican oil of 13.3 Baumé was used and the three burners each atomized over 1,800 pounds of oil per burner per hour, without smoke, carbon or other objectionable conditions. The oil burned per square foot of heating surface was 1.2 pounds per hour and per cubic foot of furnace volume 11.15 pounds per hour.

The following letter from Commander Purnell enclosed the results of the tests as given in Table II:—

"THE PEABODY ENGINEERING CORPORATION,
331 Madison Avenue,
New York, N. Y.
Attention of Mr. E. H. Peabody, President.

"GENTLEMEN:—

"I am enclosing summary of results of tests conducted on the Normand boiler, which was equipped with three Peabody air registers and three Peabody-Fisher wide-range fuel-oil atomizers.

"The Normand boiler has not been used for testing purposes for approximately five years and is not equipped with the latest appliances and instruments for securing test data. However, I believe the final calculations can be taken as representative of what can be expected from the boiler.

"While the tests were not official in so far as they were not run to meet specific requirements, the same procedure and methods of testing were followed as are customary on an official series.

"Yours very truly,

"(Signed) W. R. PURNELL,
Lieutenant Commander, U. S. N."

It is but fair to say that this rather remarkable capacity test was in a measure unexpected, as the original intention was to burn 1,500 pounds of oil per hour per burner. Also, while the boiler and furnace efficiency is low compared to results in the White-Forster tests, the results are better than the previous record on the Normand boiler as reported by Lieut. Commander (now Captain) John J. Hyland, formerly in command of the Testing Plant. Thus at the rate of .455 pound of oil per square foot of heating surface per hour the previous efficiency result was 74.5 per cent, while the present record is 80.67 per cent. No test was run previously at a rate higher than 1.05 pounds of oil per square foot of heating surface per hour, but the efficiency curve extended to the 1.2 pounds mark shows 64.5 per cent, while the present record is 68.84 per cent.

A new record in oil per burner per hour was established on September 17 by Commander Purnell when three of the six Babcock and Wilcox units were tested, the burners being supplied with larger tips. The oil used was 19.9 Baumé. The test was of one hour duration but followed

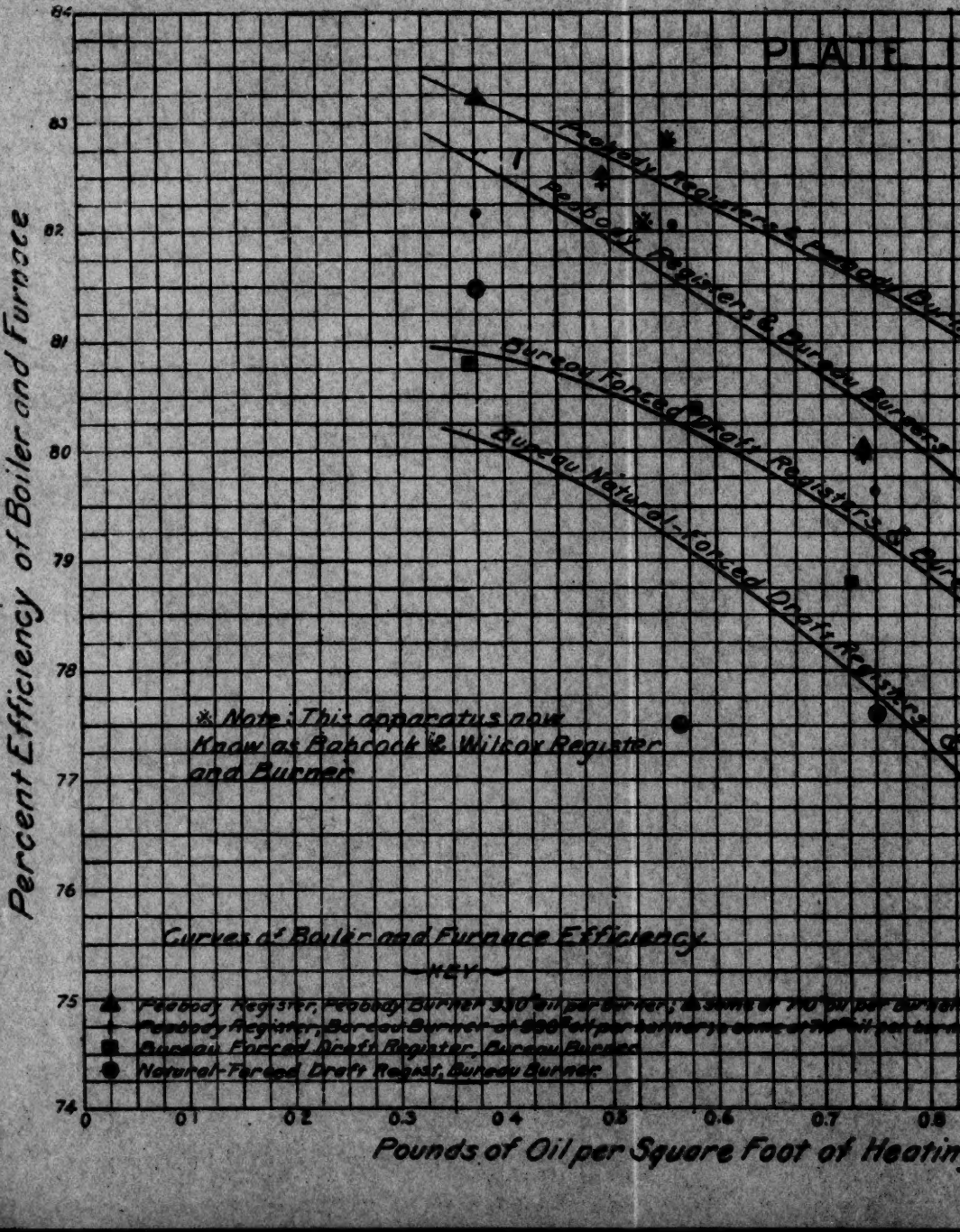
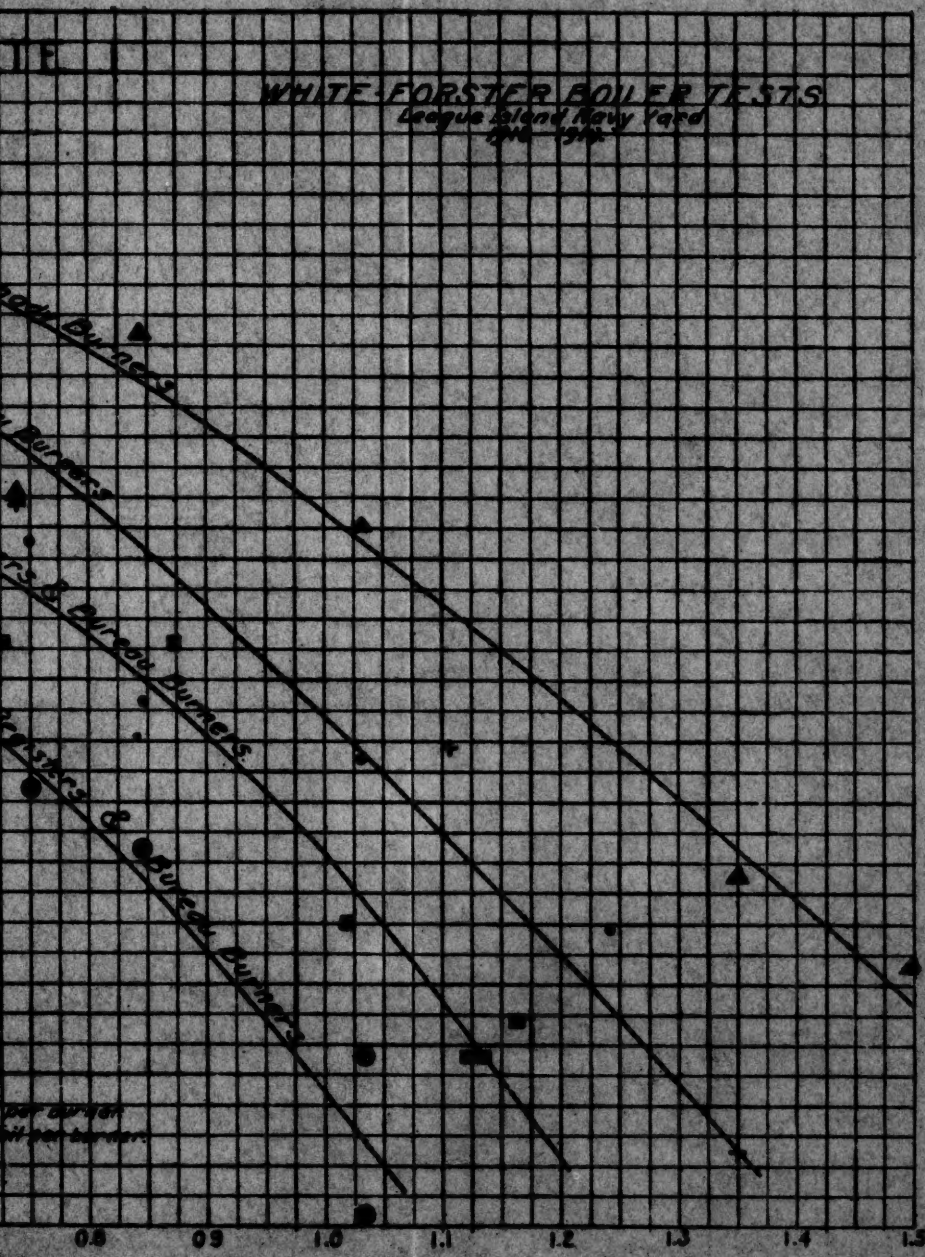


PLATE I
Percent Efficiency of Boiler and Furnace
Pounds of Oil per Square Foot of Heating
Peabody Registers & Peabody Burners
Peabody Registers & Bureau Burners
Bureau Forced Draft Registers & Burners
Bureau Natural-forced Draft Registers
* Note: This apparatus now Know as Babcock & Wilcox Register and Burner
Curves of Boiler and Furnace Efficiency
Peabody Register, Peabody Burner 3.0 Oil per Burner; Bureau Burner at 2.66 Oil per Burner
Peabody Register, Bureau Burner at 2.66 Oil per Burner
Bureau Forced Draft Register, Bureau Burner
Natural-Forced Draft Regist, Bureau Burner
84
83
82
81
80
79
78
77
76
75
74
0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8


WHITE FORSTER BOILER TESTS
League Island Navy Yard
Fuel Oil
Burners
Burners
Burners
Burners & Fuel Burners
0.8
0.9
1.0
1.1
1.2
1.3
1.4
1.5
Heating Surface per Hour.

FUEL O

EVAPORATIVE

U

& BU

Water heating surface	7565	Sq.ft.
Superheating surface	753	Sq.ft.
Total heating surface	8318	Sq.ft.
Furnace Volume	751	Cu.ft.

1	Test Number	38-2
2	Date	April 8, 1918
3	Barometer - Inches of mercury	30.06
4	Duration of test	3 hrs.
5[a]	Number of burners used	4
6[b]	Size of atomizers	14 x 24 - No3
[c]	Arrangement of burners used	8-9-10-11
6[a]	Type of air registers used	Peabody
[b]	Aggregate opening of air registers Sq. Inches	481.6
	STEAM OBSERVATIONS	
7	Steam pressure on drum - Lbs. per Sq. inch gauge	276.0
8	Steam pressure below boiler stop - Lbs per sq. inch gauge	275.8
9	Superheater temperature °F.	446.8
10	Calorimeter temperature (lower thermometer) °F.	311.9
11	Calorimeter inlet steampressure) Lbs. per sq. inch gauge	275.5
12	Feedwater temperature at boiler °F.	198.1
13	Water actually evaporated Lbs. per hour.	41735.
	FUEL OBSERVATIONS.	
14	Oil burned - Lbs. per hour	2838.7
15	Oil pressure at burners - Lbs. per sq. inch gauge	174.6
16	Oil temperature at burners - °F.	119.3
	AIR OBSERVATIONS.	
17	Dry bulb temperature - °F.	79.8
18	Wet bulb temperature °F.	60.1
19	Fireroom temperature °F.	92.3
20[a]	Stack temperature - °F.	472.8
[b]	Furnace temperature (by optical pyrometer) - °F.	2770.
21	Fireroom pressure - Inches of water	259
22	Air pressure in furnace Inches of water	.12
23	Air pressure at base of stack Inches of water	-.12
24	Smoke by Ringelmann scale	0-%
	FLUE GAS ANALYSIS	6 | 1-2 | 3
24[a]	Sampling tube numbers	
25	Carbon dioxide - %	10.47 | 10.53 | 12.23
26	Oxygen %	3.15 | 3.60 | 3.70
27	Carbon monoxide %	.00 | .00 | .00
28	Nitrogen by difference %	86.38 | 85.87 | 84.07
	ULTIMATE ANALYSIS & PHYSICAL TESTS OF FUEL	
29	Carbon %	84.88
30	Hydrogen %	12.01
31	Sulphur %	.95
32	Nitrogen %	.29
33	Oxygen and undetermined %	1.86
34	Gravity - Baumé	22.95
35	Gravity - Specific	.9162
36	Flash point - °F.	160
37[a]	Viscosity temperature °F.	120
[b]	Viscosity - Engler	3.24
38	Water & sediment %	.5
39	Calorific value B.T.U. per lb.	18938
	STEAM RESULTS	
40	Superheat - °F.	32.2
41	Quality of steam leaving drum %	99.19
42	Factor of evaporation	

EL OIL TESTING PLANT

ATIVE TEST ON WHITE FORSTER BOILER
USING PEABODY AIR REGISTERS
& BUREAU STANDARD OIL ATOMIZERS.

TABLE IA TABLE I-

Workin

32-2	32-3	32-4	32-5	32-6	32-7	32-8
April 14, 1919	April 14, 1919	April 14, 1919	April 15, 1919	April 16, 1919	April 16, 1919	May 14, 1
30.06	30.02	29.98	29.92	29.81	29.72	29.91
3 hrs.	3 hrs.	3 hrs.	3½ hrs.	3 hrs.	1½ hrs.	3 hrs.
4	6	8	9	11	11	4
⅛ x ⅛ - ⅞	⅛ x ⅛ - ⅞	⅛ x ⅛ - ⅞	⅛ x ⅛ - ⅞	⅛ x ⅛ - ⅞	⅛ x ⅛ - ⅞	⅛ x ⅛ -
9-10-11	4-6 8-9-10-11	3-4-6-7 8-9-10-11	3-4-5-6-7 8-9-10-11	All	All	9-9-10-
Peabody	Peabody	Peabody	Peabody	Peabody	Peabody	Peabo
481.6	722.4	963.2	1083.6	1324.4	1324.4	481.6
276.0	276.0	276.0	276.0	276.0	275.9	296.
275.8	271.4	266.1	262.8	253.2	242.1	286.6
246.8	460.3	472.2	473.5	470.8	467.2	460.2
311.9	312.0	310.9	310.3	297.3	292.7	311.9
275.5	273.5	272.0	270.5	267.3	263.9	294.0
198.1	198.0	198.9	198.4	198.3	198.8	194.2
7235	61949	79659	89613	107857	127560	54887
3387	42570	56763	63943	78217	94107	3720.
174.6	176.1	170.1	160.9	161.6	229.9	148.6
119.3	120.9	121.3	120.0	120.3	120.4	119.7
798	81.9	80.5	67.0	67.3	69.7	83.4
60.1	61.1	60.7	58.9	59.5	63.3	70.7
023	96.7	96.2	81.9	82.2	84.8	101.7
728	522.0	571.4	597.6	652.0	681.4	514.
770.	2950	2980.	2980	3025.	3055	2870.
2.59	3.60	4.67	5.19	6.38	7.89	4.3
.12	.39	.83	1.18	1.95	2.52	.2
.18	- .12	- .13	- .10	- .10	- .10	- .13
-¾	0	½-1	¼ - ⅜	⅜ - ¾	½ - 1	0-¾

1-2	3	6	1-2	3	6	1-2	3	6	1-2	3	6	1-2	3	6	1-2	3	11-12	3
12.55	12.23	11.46	12.25	11.63	12.63	11.80	12.16	14.65	11.50	10.86	10.44	11.43	9.60	13.40	11.85	9.96	13.19	11.10
3.60	3.70	4.57	3.89	4.10	3.03	4.67	4.10	.95	4.95	5.30	1.117	5.38	7.26	1.70	4.15	6.70	3.39	5.20
.00	.00	.00	.00	.00	.00	.00	.00	.05	.00	.00	.03	.00	.00	.00	.00	.00	.00	.00
84.07	84.07	83.97	83.96	84.27	84.48	83.53	83.74	84.44	83.65	83.84	84.53	83.15	82.14	84.90	84.00	83.26	83.45	83.70
84.09		84.09			84.09			84.90			84.90			84.90			85.4	
12.01		12.01			12.01			12.31			12.31			12.31			11.7	
.95		.95			.95			.86			.86			.86			1.16	
.29		.29			.29			.29			.29			.29			.27	
1.86		1.86			1.86			1.64			1.64			1.64			1.45	
22.95°		22.95°			22.95°			23.00°			23.00°			23.00°			24.8	
9162		.9162			.9162			.9137			.9137			.9137			.90	
160		160			160			160			160			160			210.	
120		120			120			120			120			120			104.	
3.24		3.24			3.24			3.04			3.04			3.04			4.	
5%		5%			5%			4%			4%			4%			4	
19398		18958			19182			19102			19102			19102			19371	
32.2		47.1			60.7			63.0			63.5			63.8			42.3	
99.19		99.21			99.15			99.19			98.47			98.18			99.	

FUEL OIL TESTING PLANT

EVAPORATIVE TEST ON WHITE FORSTER BOILER USING PEABODY AIR REGISTERS & BUREAU STANDARD OIL ATOMIZERS.

TABLE

	32-2	32-3	32-4	32-5	32-6	32
	April 14, 1919	April 15, 1919	April 16, 1919	April 16, 1919	April 16, 1919	Apr
	30.06	30.02	29.98	29.92	29.81	
	3 hrs.	3 hrs.	3 hrs.	3½ hrs	3 hrs	
	4	6	9	11		
	1/16 x 1/16 - 7/64	1/16 x 1/16 - 7/64	1/16 x 1/16 - 7/64	1/16 x 1/16 - 7/64	1/16 x 1/16 - 7/64	1/16 x
		4-6	3-4-6-7	3-4-5-6-7	All	
	8-9-10-11	8-9-10-11	8-9-10-11	8-9-10-11		
	Peabody	Peabody	Peabody	Peabody	Peabody	Pe
	481.6	722.4	963.2	1083.6	1924.4	13
	276.0	276.0	276.0	276.0	276.0	
guage	275.8	271.4	266.1	262.8	253.2	
	446.8	460.3	472.2	473.5	470.8	
	311.9	312.0	310.9	310.3	297.3	
ge	275.5	273.5	272.0	270.5	267.5	
	198.1	198.0	198.9	198.4	198.3	
	41735.	61949	79659	88613.	107857	12
	2839.7	4257.0	5676.3	6394.3	7821.7	
	174.6	176.1	170.1	160.9	161.6	
	119.3	120.9	121.3	120.0	120.3	
	79.8	81.9	80.5	67.0	67.3	
	60.1	61.1	60.7	58.9	59.5	
	92.3	96.7	96.2	81.9	82.2	
	472.8	522.0	571.4	597.6	652.0	6
	2770.	2950	2980.	2980	3025.	30
	259	360	467	5.19	6.30	
	.12	.39	.83	1.18	1.93	
	-.12	-.12	-.13	-.10	-.10	-
	0-¼	0	¼-½	¼-¾	¾-1	¾

	6	1-2	3	6	1-2	3	6	1-2	3	6	1-2	3	6	1-2	3	6
		*			*			*			*			*		
	12.67	12.53	12.23	4.46	12.25	11.63	12.63	11.80	12.16	14.53	11.50	10.86	10.46	11.43	9.60	13.4
	3.15	3.60	3.70	4.57	3.89	4.10	3.03	4.67	4.10	.95	4.95	5.30	1.117	5.38	7.26	1.7
	.00	.00	.00	.00	.00	.00	.00	.00	.00	.05	.00	.00	.03	.00	.00	.0
	84.18	83.87	84.07	83.97	83.86	84.27	84.34	83.53	83.74	84.45	83.55	83.84	84.37	83.19	83.14	84.9
STS OF FUEL																
	84.89			84.89			84.89			84.90			84.90			
	12.01			12.01			12.01			12.31			12.31			
	.95			.95			.95			.86			.86			
	.29			.29			.29			.29			.29			
	1.86			1.86			1.86			1.64			1.64			
	22.95°			22.95°			22.95°			23.00°			23.00°			
	.9162			.9162			.9162			.9197			.9197			
	160			160			160			160			160			
	120			120			120			120			120			
	3.24			3.24			3.24			3.04			3.04			
	5%			5%			5%			4%			4%			
	18.998			18.998			18.998			19.102			19.102			
	32.2			47.1			60.7			63.0			63.5			
	99.19			99.21			99.15			99.19			99.47			

TABLE I-A TABLE I-B

Number of tubes 4030.
Size of tubes 1" OD.
Diameter of steam drum 54"
Working pressure 300 lbs. per sq. inch gauge

-6	32-7	32-8	32-9	32-10	32-11
6, 1919	April 16, 1919	May 14, 1919	May 17, 1919	May 17, 1919	May 16, 1919
1	29.72	29.91	29.81	29.79	29.99
rs	1½ hrs.	3 hrs.	3 hrs.	2½ hrs.	4 hrs.
	11	4	6	9	11
-⅞	⅛ x ⅛ - 1/16	⅞ x ⅛ - ⅛	⅞ x ⅛ - ⅛	⅞ x ⅛ - ⅛	⅞ x ⅛ - ⅛
	All	6-9-10-11	4-6 1-2 6-9-10-11	4-5-6 6-9-10-11	All.
ody	Peabody	Peabody	Peabody	Peabody	Peabody
.4	1324.4	481.6	722.4	1083.6	1924.4
0	275.9	296.0	296.4	295.0	296.0
3.2	242.1	286.8	279.7	267.5	255.1
.8	467.2	460.2	476.5	489.8	494.9
.3	292.7	311.9	311.0	310.0	306.9
.5	263.9	294.0	292.9	286.3	273.5	
3	198.8	194.2	196.2	197.0	195.6
5 %	127560.	54887.	79018.	114831.	134239.
.7	9410.7	3720.7	5580.7	8374.4	10228.0
6	229.9	148.6	148.7	148.4	148.8
3	120.4	119.7	120.1	119.8	120.6
.3	69.7	88.4	90.8	79.8	77.3
.3	63.3	70.7	25.2	67.7	63.4
.2	84.8	101.7	99.6	90.5	85.1
0	681.4	514.	581.	673.	723.
5.	3055	2870.	3090.	3010.	3060.
2	7.83	4.353	5.976	8.085	9.543
3	2.52	.294	.852	2.138	3.293
	- .10	- .138	- .147	- .098	- .071
%	½ - 1	0 - ½	0 - ½	0 - ½	0 - ½

-2	3	6	1-2	3	11-12 10-9	3	2	11-12 10-9	6-5	3-4	11-12 10-9	6-15	2-4	11-12 10-9	6-15	2
.43	8.60	13.40	11.85	9.96	13.19	11.10	14.00	12.70	14.03	11.25	13.10	13.57	12.85	12.51	13.38	13.61
.38	7.86	1.70	4.15	6.70	3.39	5.20	1.60	2.71	1.75	4.82	2.74	2.55	3.47	2.42	2.40	1.95
0	.00	.00	.00	.00	.00	.00	.00	.00	.00	.01	.12	.00	.00	.24	.40	.00
8	83.14	84.90	84.00	83.36	83.26	83.70	84.46	83.53	84.20	83.32	84.40	83.70	83.60	83.36	83.87	84.58
90		84.90		85.42				85.36			85.36			85.50		
31		12.31		11.73				11.76			11.76			11.79		
86		.86		1.16				.98			.98			1.18		
29		.29		.24				.25			.25			.18		
44		1.64		1.45				1.65			1.65			1.35		
00°		23.00°		24.85				25.5			25.5			25.8		
9137		.9137		.9051				.9014			.9014			.8985		
0		160		210.				255			255			245		
0		120		104.				104.			104.			104.		
04		3.04		6.54				4.07			4.07			4.00		
%		.4%		.4%				.2%			.2%			.2%		
2.		19102.		19216.				19166.			19160.			19256.		
.5		63.8		42.3				60.7			77.8			86.9		
21		98.18		99.07				98.04			98.01			98.25		

		6	1-2	3
	FLUE GAS ANALYSIS			
24	Sampling tube numbers			
25	Carbon dioxide - %	12.67	12.53	12.23
26	Oxygen %	3.15	3.40	3.70
27	Carbon monoxide %	.00	.00	.00
28	Nitrogen by difference %	84.18	84.07	84.07
	ULTIMATE ANALYSIS & PHYSICAL TESTS OF FUEL			
29	Carbon %	84.69		
30	Hydrogen %	12.01		
31	Sulphur %	.95		
32	Nitrogen %	.29		
33	Oxygen and undetermined %	1.86		
34	Gravity - Baume	22.95		
35	Gravity - Specific	.9162		
36	Flash point - °F.	160		
37a	Viscosity temperature °F.	120		
b	Viscosity - Engler	3.24		
38	Water & sediment %	5%		
39	Colorific value B.T.U. per lb.	18958		
	STEAM RESULTS			
40	Superheat - °F	32.2		
41	Quality of steam leaving drum %	99.19		
42	Factor of evaporation	1.0944		
43	Actual evaporation per lb. of fuel - lbs.	14.697		
44	Equivalent evaporation per lb. of fuel lbs.	16.084		
45	Actual evaporation per sq. ft. of water H.S. lbs. per hr.	5.517		
46	Equivalent evaporation per sq. ft. water H.S. lbs. per hour	6.039		
47	% of total heat transfer taking place in superheaters	2.92%		
	OIL RESULTS			
48	Oil burned per sq. ft. of water H.S. lbs. per hour	.375		
49	Oil burned per cu. ft. of furnace volume lbs. per hr.	.3.781		
50	Oil burned per burner per hour lbs.	708.9		
	AIR RESULTS			
51	Relative Humidity %	29.2		
52	Dew Point - °F.	44.4		
53a	Moisture - lbs. per cu. ft.	.00048		
b	Vapor pressure (e) - Inches of Mercury.	.291		
54	Weight of dry air - lbs. per cu. ft.	.07197		
55	Weight of air at atmospheric conditions - lbs. per cu. ft.	.07245		
56	Air at atmospheric conditions theoretically required for combustion of one lb of fuel - cu. ft.	183.6		
	RESULTS FROM ORSATT & ULTIMATE ANALYSIS			
57	Weight of dry gasses per lb. of fuel - lbs.	17.02		
58	Weight of dry air per lb. of fuel - lbs.	17.22		
59	Excess air from Orsatt - %	19.7		
60	Volume of air at atmospheric conditions used per lb. of fuel - Cu. ft.	—		
61	Volume of air at fireroom conditions used per lb. of fuel - Cu. ft.	238.3		
62	Velocity of air through vanes of air registers Ft. per minute.	3398		
63	% excess air over theoretical amount required for complete combustion	23.6		
	HEAT BALANCE	B. T. U.	%	
64	Heat absorbed by boiler	15608	82.16	
65	Loss by imperfect combustion	0	—	
66	Loss with dry chimney gases	1554	8.19	
67a	Loss with moisture from hydrogen in fuel	1211	6.38	
b	Loss with moisture in fuel	—	—	
c	Loss with moisture in air	21.	.11	
68	Radiation and error	564	2.96	
69	Total	18958	100.00	
70	Boiler and furnace efficiency %	82.16		

Remarks - Readings used in calculations.

1-2	3	6	1-2	3	6	1-2	3	6	1-2	3	6	1-2	3	6	1-2	3	6	1-2	3
10.55	12.23	11.46	12.25	12.63	12.65	11.80	13.16	14.55	11.60	10.96	14.44	11.43	9.60	12.40	11.85	9.96	13.13	11.10	10
.360	.370	3.57	3.89	4.10	3.03	4.67	4.10	.95	4.85	5.30	1.117	5.38	7.26	1.70	4.15	6.70	3.39	5.20	1
.00	.00	.00	.00	.00	.00	.00	.00	.05	.00	.00	.03	.00	.00	.00	.00	.00	.00	.00	
83.67	84.61	83.97	80.86	86.57	84.44	83.53	80.24	84.45	83.55	82.84	84.54	83.43	81.14	84.90	84.01	83.36	83.43	83.70	8
84.89		84.89		84.89		84.90		84.90		84.90								85.42	
12.01		12.01		12.01		12.31		12.31		12.31								11.73	
.95		.95		.95		.86		.86		.86								1.16	
.29		.29		.29		.29		.29		.29								.24	
1.86		1.86		1.86		1.64		1.64		1.64								1.43	
22.95°		22.95°		22.95°		23.00°		23.00°		23.00°								24.85	
.9162		.9162		.9162		.9137		.9137		.9137								.905	
160		160		160		160		160		160								210.	
120		120		120		120		120		120								104.	
3.24		3.24		3.24		3.04		3.04		3.04								6.54	
3%		3%		3%		4%		4%		4%								4%	
18998		18998		18998		19102		19102		19102								19216	
32.2		47.1		60.7		63.0		63.5		63.8								42.3	
99.19		99.21		99.15		99.13		99.41		98.18								99.07	
0944		1.1021		1.1109		1.1184		1.1119		1.1105								1.101	
4.697		14.552		14.034		13.858		13.790		13.555								14.475	
6.004		16.067		15.590		16.416		15.333		15.053								16.328	
5.517		8.189		10.530		11.714		14.257		16.862								7.255	
6.039		9.042		11.698		13.031		16.052		18.725								8.027	
2.92%		3.75%		4.47%		4.59%		5.10%		5.22%								3.61%	
.375		.563		.750		.845		1.034		1.244								.491	
3.781		5.668		7.558		8.514		10.415		12.531								4.934	
708.9		709.5		709.5		710.6		711.1		855.5								930.2	
29.2		27.5		31.9		60.3		62.8		70.4								31.1	
44.4		45.4		45.6		53.0		54.7		59.8								59.0	
00048		.00050		.00050		.00065		.00069		.00082								.00	
.291		.303		.305		.402		.428		.513								.49	
07197		.07145		.07158		.07320		.07307		.07255								.07	
07245		.07195		.07208		.07385		.07374		.07337								.071	
193.6		195.0		194.3		191.4		191.8		193.1								197.9	
17.02		17.39		18.03		18.48		18.60		17.96								16.3	
17.22		17.61		18.21		18.69		18.72		18.24								16.37	
19.7		21.6		27.3		29.4		32.9		33.3								18.4	
—		—		—		—		—		—								—	
238.9		246.5		254.4		255.3		256.2		261.4								233.1	
3396		3498		3610		3620		3644		4301								4337	
23.6		26.4		30.9		33.4		33.6		30.2								17.8	
B.T.U.	%	B.T.U.	%	B.T.U.	%	B.T.U.	%	B.T.U.	%	B.T.U.	%	B.T.U.	%	B.T.U.	%	B.T.U.	%	B.T.U.	%
15600	89.16	15591	89.07	15129	78.68	14960	78.32	14878	77.89	14607	76.47	16832							
0	—	0	—	0	—	30	.16	18	.09	0	—	0							
554	3.18	1775	9.34	2056	10.69	2287	11.97	2524	13.22	2572	13.47	1606							
911	5.20	1291	7.01	1357	7.14	1420	7.43	1440	7.54	1460	7.64	1291							
—	—	—	—	—	—	—	—	—	—	—	—	—							
21	.11	25	.13	28	.16	40	.21	47	.25	46	.30	36							
504	2.65	276	1.45	420	2.25	365	1.91	166	.87	405	2.12	444							
988	max	18998	max	18998	max	19102	max	19102	max	19102	max	19216							
82.16		82.07		79.64		78.32		77.89		76.47		82.4							

	6	1-2	3	6	1-2	3	6	1-2	3	6	1-2	3	6
	12.67	12.55	12.23	11.46	12.25	11.63	12.53	11.00	12.16	14.65	11.50	10.96	11.46	11.43	8.60	13.40
	3.15	3.60	3.70	3.57	3.89	4.10	3.03	4.67	4.10	.95	4.85	5.30	1.117	5.38	7.26	1.70
	.00	.00	.00	.00	.00	.00	.00	.00	.00	.05	.00	.00	.02	.00	.00	.00
	84.18	83.85	84.07	83.97	83.86	84.27	84.44	83.33	83.74	84.35	83.65	83.04	84.36	83.19	84.14	84.90
TS OF FUEL																
	84.69			84.69			84.89			84.90			84.90			
	12.01			12.01			12.01			12.31			12.31			
	.95			.95			.95			.86			.86			
	.29			.29			.29			.29			.29			
	1.86			1.86			1.86			1.64			1.64			
	22.95°			22.95°			22.95°			23.00°			23.00°			
	.9162			.9162			.9162			.9137			.9137			
	160			160			160			160			160			
	120			120			120			120			120			
	3.24			3.24			3.24			3.04			3.04			
	5%			5%			5%			4%			4%			
	18998			18998			18998			19102			19102			
	32.2			47.1			60.7			63.0			63.5			
	99.19			99.21			99.15			99.13			98.41			
	1.0944			1.1021			1.1109			1.1184			1.1119			
	14.697			14.552			14.034			13.858			13.790			
	16.084			16.067			15.590			16.416			15.333			
	5.517			8.189			10.530			11.714			14.287			
our	6.039			9.042			11.698			13.031			16.852			
	2.92%			3.75%			4.47%			4.59%			5.10%			
	.375			.563			.750			.845			1.034			
	3.781			5.668			7.558			8.514			10.415			
	709.9			709.5			709.5			710.6			711.1			
	29.2			27.5			31.9			60.3			62.8			
	44.4			45.4			45.6			53.0			54.7			
	.00048			.00050			.00050			.00065			.00069			
	.291			.303			.305			.402			.428			
	.07197			.07145			.07158			.07320			.07307			
u.ft.	.07245			.07195			.07206			.07385			.07376			
d	193.6			195.0			194.3			191.4			191.8			
ANALYSIS																
	17.02			17.39			18.03			18.48			18.60			
	17.22			17.61			18.21			18.69			18.72			
	19.7			21.6			27.3			29.4			32.9			
of fuel-Cu.ft.	—			—			—			—			—			
fuel-Cu.ft.	239.9			246.5			254.4			255.3			256.2			
minute.	3399.			3498			3610			3628.			3644.			
complete combustion	23.6			26.4			30.9			33.4			33.6			
	B.T.U.	%		B.T.U.	%		B.T.U.	%		B.T.U.	%		B.T.U.	%		B.
	15609	82.16		15591	82.07		15129	79.64		14960	78.32		14830	77.69		14
	0	—		0	—		0	—		30	.16		10	.05		
	1554	8.18		1775	9.34		2056	10.82		2287	11.97		2542	13.32		2
	1311	6.90		1331	7.01		1357	7.14		1420	7.43		1420	7.44		1
	—	—		—	—		—	—		—	—		—	—		
	21.	.11		25	.13		28	.15		40	.21		47	.25		
	504	2.65		276	1.45		420	2.25		365	1.91		166	.87		4
	18998	100		18998	100		18998	100		19102	100		19102	100		19
	82.16			82.07			79.64			78.32			77.69			

as.

2	3	6	1-2	3	11-10	3	2	6-12	6-5	2-4	11-12	6-15	2-4	11-12	6-15	2			
					10-9			10-9			10-9			10-9					
3	9.60	14.40	11.85	9.96	13.13	11.10	14.00	12.70	14.03	11.25	13.10	13.57	12.84	9.51	13.38	13.81			
	7.26	1.70	4.15	6.70	5.39	5.20	1.60	3.71	1.75	4.82	2.74	2.55	5.47	2.48	3.46	1.95			
	.00	.00	.00	.00	.00	.00	.00	.00	.00	.01	.12	.00	.05	.24	.40	.00			
	82.14	84.90	84.00	83.34	83.24	83.70	84.46	83.59	84.78	83.98	84.64	83.92	83.60	83.85	83.78	84.56			
90		84.90		85.42				85.36			85.96			85.50					
11		12.31		11.73				11.76			11.76			11.79					
16		.86		1.16				.98			.98			1.18					
29		.29		.24				.25			.25			.18					
4		1.64		1.45				1.65			1.65			1.35					
00°		23.00°		24.85				25.5			25.5			25.8					
137		.9137		.9051				.9014			.9014			.8995					
0		160		210.				255			255			245					
0		120		104.				104.			104.			104.					
04		3.04		6.54				4.07			4.07			4.00					
%		.4%		.4%				.2%			.2%			.2%					
2.		19102.		19216.				19166.			19166.			19256.					
5		63.8		42.3				60.7			77.8			86.9					
1		98.18		99.07				99.04			99.01			98.85					
119		1.1105		1.1064				1.1149			1.1229			1.1281					
90		13.555		14.752				14.159			13.712			13.125					
33		15.053		16.322				15.786			15.397			14.806					
87		14.862		7.255				10.445			15.179			17.745					
52		18.725		8.027				11.645			17.044			20.019					
%		5.227		3.61%				4.37%			5.36%			5.80%					
34		1.244		.492				.738			1.107			1.352					
15		12.531		4.954				7.431			11.151			13.619					
		855.5		930.2				930.1			930.5			929.8					
.8		70.4		31.1				48.1			49.5			45.1					
7		59.8		59.0				68.6			59.0			54.5					
00069		.00042		.00080				.00110			.00080			.00069					
428		.513		.499				.698			.499			.424					
07307		.07255		.07022				.07006			.07201			.07367					
07376		.07337		.07102				.07116			.07281			.07436					
8		193.1		197.9				196.1			192.7			189.1					
60		17.96		16.32				16.90			16.25			15.68					
72		18.24		16.37				17.02			16.44			15.80					
9		23.3		18.4				20.5			14.3			12.4					
3		251.4		233.1				242.9			228.3			214.5					
4		4301.		4337.				4518.			4249.			3989.					
6		30.2		17.8				22.6			14.3			11.3					
U.	%	B.T.U	%	B.T.U.	%		B.T.U	%	B.T.U.	%	B.T.U.	%							
8.	mois	14607	79.91	15832.	82.85		15319.	79.93	14941.	77.96	14368	74.62							
	.69	0	—	0.	—		0.	—	72.	.41	152.	.79							
44	12.26	2572	13.47	1606.	8.34		1946.	10.15	2263.	11.81	2389.	12.40							
0	2.69	1460	7.66	1291.	6.78		1389.	7.24	1305	7.23	1412.	7.37							
—	—	—	—	—	—		—	—	—	—	—	—							
	.23	58.	.30	36.	.18		60.	.31	50.	.26	44.	.23							
	.87	405	2.12	444.	2.30		512.	2.67	448	2.36	684.	4.52							
2	mois	19102	mois	19216.	mois		19166.	mois	19166.	mois	19256.	mois							
7.89		76.47		82.43			79.93		77.96		74.62								

Water heating surface	7565	Sq. ft.
Superheating surface	753	Sq. ft.
Total heating surface	8318	Sq. ft.
Furnace volume	751	Cu. ft.

FUEL OIL TES

EVAPORATIVE TEST ON WA

USING PEABODY AIR REG

& PEABODY OIL ATOMIZ

1	Test Number.	30-14
2	Date.	June 9 1914
3	Barometer - Inches of mercury.	30.29
4	Duration of test.	5 hrs.
5a	Number of burners used.	4
b	Size of atomizers.	#104
c	Arrangement of burners used.	8-9-10-11
6a	Type of air registers used.	Peabody
b	Aggregate opening of air registers - Sq. inches	491.6
	STEAM OBSERVATIONS.	
7	Steam pressure on drum - Lbs. per sq. inch, gauge.	296.1
8	Steam pressure below boiler stop - Lbs. per sq. inch, gauge.	290.1
9	Superheater temperature °F.	466.3
10	Calorimeter temperature (lower thermometer) °F	315.6
11	Calorimeter (inlet steam pressure) Lbs. per sq. inch gauge.	294.1
12	Feedwater temperature at boiler - °F.	197.6
13,	Water actually evaporated - Lbs. per hour.	55525.
	FUEL OBSERVATIONS.	
14	Oil burned - Lbs. per hour.	3786.
15	Oil pressure at burners - Lbs. per sq. inch, gauge.	197.2
16	Oil temperature at burners - °F.	119.6
	AIR OBSERVATIONS.	
17	Dry bulb temperature - °F.	69.6
18	Wet bulb temperature - °F.	72.9
19	Fireroom temperature - °F.	101.7
20a	Stack temperature - °F.	532.
b	Furnace temperature (by optical pyrometer) - °F.	———
21	Fireroom pressure - Inches of water.	4.037
22	Air pressure in furnace - Inches of water.	.370
23	Air pressure at base of stack - Inches of water.	−.093
24	Smoke by Ringelmann scale.	0 - ½

	FLUE GAS ANALYSIS	12-16 No.	2	6
24½	Sampling tube numbers.			
25	Carbon dioxide - %	13.06	12.40	11.79
26	Oxygen - %	3.27	3.50	4.68
27	Carbon monoxide - %	.11	.01	.04
28	Nitrogen by difference - %	83.56	84.09	83.49

	ULTIMATE ANALYSIS & PHYSICAL TESTS OF FUEL.	
29	Carbon - %	86.00
30	Hydrogen - %	11.70
31	Sulphur - %	.65
32	Nitrogen - %	.24
33	Oxygen and undetermined - %	1.41
34	Gravity - Baumé	26.1°
35	Gravity - Specific.	.8978
36	Flash point - °F.	210.
37a	Viscosity temperature - °F.	120.
b	Viscosity - Engler.	3.16
38	Water & sediment - %	.3%
39.	Calorific value - B.T.U. per lb.	19300.
	STEAM RESULTS.	
40	Superheat - °F.	46.3
41	Quality of steam leaving drum - %	99.28
42	Factor of evaporation.	1.1046
43	Actual evaporation per lb. of fuel	

Table

L TESTING PLANT

r on WHITE FORSTER BOILER

Y AIR REGISTERS.

% ATOMIZERS.

Number of tubes 4032
Size of tubes 1" O.D.
Diameter of steam drum 64"
Working pressure 300 Lbs. per s

-14	30-15	30-16	30-17	30-18	30-19	30-20.
9.1919	June 9.1919	June 10.1919.	June 10.1919	June 13.1919	June 13.1919	June 13.1919
.29	30.28	30.32	30.28	30.40	30.32	30.24
hrs.	3 hrs.	3 hrs.	2 hrs.	3 hrs.	2½ hrs.	2 hrs.
	6	11	11	4	9	11
4	#104	#104	#104	#103	#103	#103.
10-11	4—6 8-9-10-11	All.	All.	8-9-10-11	4 1 2 3 6 8-9-10-11	All.
body.	Peabody.	Peabody.	Peabody.	Peabody.	Peabody.	Peabody.
1.6	728.4	1324.4	1324.2	491.6	1003.6	1324.4
6.1	295.8	295.2	295.6	297.1	296.0	295.9
0.1	285.5	294.2	285.5	294.9	278.3	292.9
.53	480.6	486.5	484.5	459.5	482.9	483.6
5.5	314.7	302.4	296.7	315.3	—	—
6.1	292.3	283.2	281.6	296.1	—	—
7.6	196.9	196.9	198.0	195.1	194.7	197.2
23.	796.32.	1394.52.	1382.67.	42.623.	982.08	1106.63.
66.	5584.	10231.	11355.	2889.	6590.	7810.
7.2	197.6	197.3	245.9	169.4	170.6	169.3
9.6	119.6	120.0	120.0	119.9	120.0	120.0
2.8	88.1	86.5	90.5	106.5	101.9	100.3
3.0	73.0	76.5	76.8	77.6	71.7	70.9
1.7	99.3	97.8	109.1	115.1	115.5	114.6
2.	614.	715.	789.	490.	617	649.
—	3120.	3175.	3170.	2850.	3090.	3100.
037	6.987	8.762	9.648	2.739	4.952	5.827
170	1.011	3.306	3.707	.116	1.186	1.881
93	-.046	+.192	+.848	-.115	-.035	+.046
-14	0-3/4	1½-3/4	1½-2	0-1/4	0-1/4	0-1/4

2	6	12-18 Ht.	2	6	12-18 Ht.	2	6	12-18 Ht.	2	6	12-18 Ht.	2	6	12-18 Ht.	2	6	12-18 Ht.	2	6
2.40	1.473	12.07	11.90	11.85	12.74	12.60	12.03	13.01	12.60	12.38	12.50	12.43	12.80	12.66	12.78	12.10	12.60	11.66	12.60
3.40	4.78	4.70	4.45	4.87	3.71	3.46	2.72	3.38	3.46	2.67	3.68	1.78	2.86	4.81	4.87	3.46	4.46	4.84	4.87
.05	.06	.02	.00	.06	.00	.16	.02	.02	.18	.06	.04	.07	.00	.02	.00	.02	.00	.10	.08
1.00	83.70	89.2	89.67	89.14	88.21	88.03	89.27	89.58	87.84	87.04	88.21	89.78	89.15	87.36	87.46	87.96	88.30	88.22	87.66
.00		86.00			86.90			86.90			86.90			86.90			86.90		
.70		11.70			11.60			11.60			11.60			11.60			11.60		
.65		.65			.65			.65			.86			.86			.86		
.84		.84			.84			.84			.89			.89			.89		
41		1.41			1.61			1.61			1.45			1.45			1.45		
.1°		26.1°			25.55°			25.55°			25.55°			25.55°			25.65°		
978		.8978			.9010			.9010			.9010			.9010			.9010		
10.		210.			195.			195.			215			215			216		
10.		120.			120.			120.			120.			120.			120.		
16		3.16			3.06			3.06			3.17			3.17			3.17		
%		.3%			.3%			.3%			.8%			.8%			.8%		
0.		19300.			19300.			19300.			19280.			19280.			19280.		
6.3		68.9			78.7			80.6			39.3			67.4			69.8		
2.8		99.26			98.60			98.29			99.26			—			—		
65%		1.1161			1.1222			1.1206			1.1041			1.1204			1.1167		

7565 Sq. ft.
753 Sq. ft.
8318 Sq. ft.
751 Cu. ft.

FUEL OIL TESTING PLANT

EVAPORATIVE TEST ON WHITE FORSTER BOILER

USING PEABODY AIR REGISTERS.
& PEABODY OIL ATOMIZERS.

	30-14	30-15	30-16
	June 9 1919	June 9 1919	June 10 1919
ercury	30.29	30.28	30.32
	3 hrs.	3 hrs.	3 hrs.
d.	4	6	11
	#104	#104	#104
		4—6	All.
s used.	8-9-10-11	8-9-10-11	
ed.	Peabody	Peabody	Peabody
ir registers - Sq. inches	481.6	728.4	1324.4
EAM OBSERVATIONS.			
um - Lbs. per sq. inch, gauge.	296.1	295.8	295.2
boiler stop - Lbs. per sq. inch, gauge.	292.1	285.5	284.2
rer °F.	465.3	480.6	486.6
(lower thermometer) °F	315.6	314.7	302.4
ressure) Lbs. per sq. inch gauge.	294.1	292.3	283.2
e at boiler - °F.	197.6	196.9	196.9
ted - Lbs. per hour.	55323.	79632.	139452.
L OBSERVATIONS.			
our.	3786.	5584.	10231.
rs - Lbs. per sq. inch, gauge.	197.2	197.6	197.3
here - °F.	119.6	119.6	120.0
OBSERVATIONS.			
- °F.	82.8	88.1	84.6
°F.	73.9	73.0	74.5
°F.	101.7	99.3	97.8
	532.	614.	715.
by optical pyrometer) - °F.	—	3120.	3175.
es of water.	4.037	6.987	8.762
- Inches of water.	.370	1.011	3.306
stack - Inches of water.	-.093	-.046	+.198
n scale.	0-1/4	0-3/4	1/2-3/4

E GAS ANALYSIS	12-18	2	6	12-18	2	6	12-18	2	6	12-18
	19 °			19 °			19 °			19 °
	13.08	12.40	11.72	12.07	11.92	11.95	12.78	12.60	12.40	13.00
	3.27	3.90	4.58	4.70	4.43	4.87	3.71	3.40	4.22	3.58
	.11	.05	.04	.02	.00	.06	.00	.16	.04	.02
- %	83.54	84.05	83.76	83.21	83.65	83.14	83.51	83.84	84.34	83.40
LYSIS & PHYSICAL TESTS OF FUEL.										
	86.00			86.00			86.90			
	11.70			11.70			11.60			
	.65			.65			.65			
	.24			.24			.24			
ed - %	1.41			1.41			1.61			
	26.1°			26.1°			25.65°			
	.8978			.8978			.9010			
	210.			210.			196.			
	120.			120.			120.			
	3.16			3.16			3.06			
	.37			.32			.37			
per lb.	19300.			19300.			19300.			
STEAM RESULTS.										
	46.3			68.9			78.7			
um - %	99.28			99.26			98.60			
	1.1065			1.1161			1.1222			

Table Ic

NT
OILER

Number of tubes 4032
Size of tubes 1" O.D.
Diameter of steam drum 54"
Working pressure 300 lbs. per sq. inch gauge.

16	30-17	30-18	30-19	30-20.	
1919.	June 10. 1919.	June 13. 1919	June 13. 1919	June 13. 1919.	
92.	30.38	30.40	30.38	30.26	
3.	2 hrs.	3 hrs.	2½ hrs.	2 hrs.	
	11	4	9	11	
	#104	#103	#103	#103.	
	All.	8-9-10-11	4-1-2-3-6 8-9-10-11	All.	
ody.	Flinbody	Flinbody	Flinbody	Flinbody.	
	138.47	491.6	1083.6	1324.4	
2	295.6	297.1	296.0	295.9	
2	245.5	294.9	278.3	272.9	
5	494.5	459.5	482.9	483.6	
4	296.7	315.3	—	—	
2	281.6	296.1	—	—	
9	190.0.	195.1	194.7	197.2	
2.	153467.	42.623.	92208	110663.	
	11355.	2859.	6390.	7810.	
3	245.9	162.4	170.6	169.3	
0	120.0	119.9	120.0	120.0	
T	90.5	104.5	101.9	100.3	
	76.8	77.6	71.7	70.9	
	109.1	115.1.	115.5	114.6	
	789.	490.	617	649.	
	3170.	2880.	3090	3100.	
2	2.544	2.739	4.952	5.827	
6	3.707	.116	1.180	1.881	
	+.842	-.115	-.035	+.046	
	1½-8	0-½	0-½	0-½	

	6	12.56	2	6	12.56	2	6	12.56	2	6	12.56	2	6	12.56	2	6												
		18.00	13.00	18.00	13.00	18.00	13.00	18.00	18.00	18.00	18.00	18.00	18.00	18.00	11.00	18.00												
		3.00	3.00	3.00	2.67	3.00	1.33	3.00	3.00	3.00	3.00	3.00	3.00	3.00	3.00	3.00												
		.00	.09	.18	.06	.00	.00	.00	.00	.00	.00	.00	.00	.00	.10	.05												
		84.51	88.05	84.51	84.51	88.05	88.05	88.05	88.05	84.51	84.51	84.51	84.51	88.05	88.05	84.51												

00	8490	8490	8490	8490	
50	11.60	11.40	11.40	11.40	
45	.65	.96	.96	.96	
24	.24	.89	.89	.89	
1	1.61	1.45	1.45	1.45	
55°	25.65°	25.65°	25.65°	25.65°	
10	.9010	.9010	.9010	.9010	
	195.	215	215	215	
	120.	120.	120	120.	
	306	3.17	3.17	3.17	
2	32	.32	.32	.32	
	19300.	19280.	19280.	19280.	
7	80.6	39.3	67.4	69.8	
50	98.29	99.84	—	—	
2	1.1806	1.1041	1.1806	1.1147	

24a	Sampling tube numbers.	No.	2	3
25	Carbon dioxide - %	13.6	12.40	11.72
26	Oxygen - %	3.27	3.80	4.98
27	Carbon monoxide - %	.11	.05	.06
28	Nitrogen by difference - %	83.06	83.75	83.24

ULTIMATE ANALYSIS & PHYSICAL TESTS OF FUEL.

29	Carbon - %	86.00.
30	Hydrogen - %	11.70
31	Sulphur - %	.65
32	Nitrogen - %	.24
33	Oxygen and undetermined - %	1.41
34	Gravity - Baumé	26.1°
35	Gravity - Specific.	.8978
36	Flash point - °F.	210.
37a	Viscosity temperature - °F.	120.
b	Viscosity - Engler.	3.16
38	Water & sediment - %	.3%
39	Calorific value - B.T.U. per lb.	19300.

STEAM RESULTS.

40	Superheat - °F.	46.3
41	Quality of steam leaving drum - %	99.28
42	Factor of evaporation.	1.1056
43	Actual evaporation per lb. of fuel - Lbs.	14.849
44	Equivalent evaporation per lb. of fuel - Lbs.	16.416
45	Actual evaporation per sq. ft. of water H.S. - Lbs. per hour.	7.313
46	Equivalent evaporation per sq. ft. of water H.S. - Lbs. per hour.	8.085
47	% of total heat transfer taking place in superheaters.	3.70 %

OIL RESULTS

48	Oil burned per sq. ft. of water H.S. - Lbs. per hour.	.493
49	Oil burned per cu. ft. of furnace volume - Lbs. per hour.	4.961
50	Oil burned per burner per hour - Lbs.	991.5

AIR RESULTS.

51	Relative Humidity - %	470
52	Dew Point - °F.	65A
53a	Moisture - Lbs. per cu. ft.	.001003
b	Vapor pressure (e) : Inches of mercury.	.634
54	Weight of dry air - Lbs. per cu. ft.	.071883
55	Weight of air at atmospheric conditions - Lbs. per cu. ft.	.072986
56	Air at atmospheric conditions theoretically required for combustion of one lb. of fuel. Cu. ft.	193.6

RESULTS FROM ORSATT & ULTIMATE ANALYSIS.

57	Weight of dry gases per lb. of fuel - Lbs.	16.49
58	Weight of dry air per lb. of fuel - Lbs.	16.59
59	Excess air from Orsatt - %	17.6
60	Volume of air at atmospheric conditions used per lb. of fuel - Cu. ft.	—
61	Volume of air at firerom conditions used per lb. of fuel - Cu. ft.	2306
62	Velocity of air through vanes of air registers - Ft. per minute.	4294.
63	% excess air over theoretical amount required for complete combustion.	191

HEAT BALANCE.

		B.T.U.	%
64	Heat absorbed by boiler	15930	82.54
65	Loss by imperfect combustion	73	.38
66	Loss with dry chimney gases.	1703	8.82
67a	Loss with moisture from hydrogen in fuel	1296	6.72
b	Loss with moisture in fuel.	—	—
c	Loss with moisture in air	47	.24
68	Radiation and error -	251	1.30
69	Total	19300.	100.00
70	Boiler and furnace efficiency - %	82.54	

REMARKS * Readings used in calculations.

	3.27	3.50	4.98		4.70	4.45	4.47	3.71	3.45	3.33
%	.11	.09	.04		.08	.00	.06	.00	.10	.0
ince - %	93.60	84.05	83.70		93.91	89.67	89.12	89.81	89.90	89.67
ANALYSIS & PHYSICAL TESTS OF FUEL.										
	86.00				86.00			86.90		
	11.70				11.70			11.60		
	.65				.65			.65		
	.84				.84			.84		
rmined - %	1.41				1.41			1.41		
	26.1°				26.1°			25.55°		
	.8978				.8978			.9010		
	210.				210.			196.		
- °F.	180.				180.			180.		
	3.16				3.16			3.06		
%	.3%				.3%			.3%		
T.U. per lb.	19300.				19300.			19300.		
STEAM RESULTS.										
	46.3				68.9			78.7		
ing drum - %	99.28				99.26			98.60		
n.	1.1056				1.1161			1.1222		
per lb. of fuel - Lbs.	14.848				14.261			13.630		
ion per lb. of fuel - Lbs.	16.416				15.917			15.296		
per sq. ft. of water H.S. - Lbs. per hour.	7.313				10.526			10.434		
per sq. ft. of water H.S. - Lbs. per hour.	8.085				11.748			20.687		
fer taking place in superheaters.	3.70%				4.57%			5.62%		
OIL RESULTS										
ft. of water H.S. - Lbs. per hour.	.493				.738			1.352		
of furnace volume - Lbs. per hour.	4.961				7.435			13.623		
r per hour - Lbs.	931.5				930.7			930.1		
AIR RESULTS.										
%	47.0				47.3			36.8		
	65.8				65.7			69.6		
cu. ft.	.001003				.001000			.001130		
): Inches of mercury.	.634				.638			.720		
- Lbs. per cu. ft.	.071883				.072688			.078239		
atmospheric conditions - Lbs. per cu. ft.	.072986				.073688			.079369		
c conditions theoretically required for										
e lb. of fuel. Cu. ft.	193.6				191.6			169.4		
S FROM ORSATT & ULTIMATE ANALYSIS.										
ases per lb. of fuel - Lbs.	16.69				17.85			16.94		
per lb. of fuel - Lbs.	16.59				17.94			17.04		
n Orsatt - %	17.6				27.6			20.5		
atmospheric conditions used per lb. of fuel - Cu. ft.	—				—			—		
fire room conditions used per lb. of fuel - Cu. ft.	2305				2470			2327		
ough vents of air registers - ft. per minute.	4894.				4598.			4329.		
er theoretical amount required for complete combustion.	19.1				28.8			22.9		
HEAT BALANCE.	B.T.U.	%		B.T.U.	%		B.T.U.	%	
y boiler	15930	82.54		14916	80.08		14843	76.91	
f combustion	73	.38		14	.07		0	0	
himney gases.	1703	8.82		2205	11.83		2407	12.47	
ure from hydrogen in fuel	1296	6.71		1399	6.84		1379	7.14	
ure in fuel.	—	—		—	—		—	—	
re in air	47	.24		60	.31		76	.39	
rror -	251	1.30		286	1.53		493	2.55	
	19300.	100.00		19300	100.00		19300.	100.00	
ace efficiency - %	82.54			80.08			76.91		

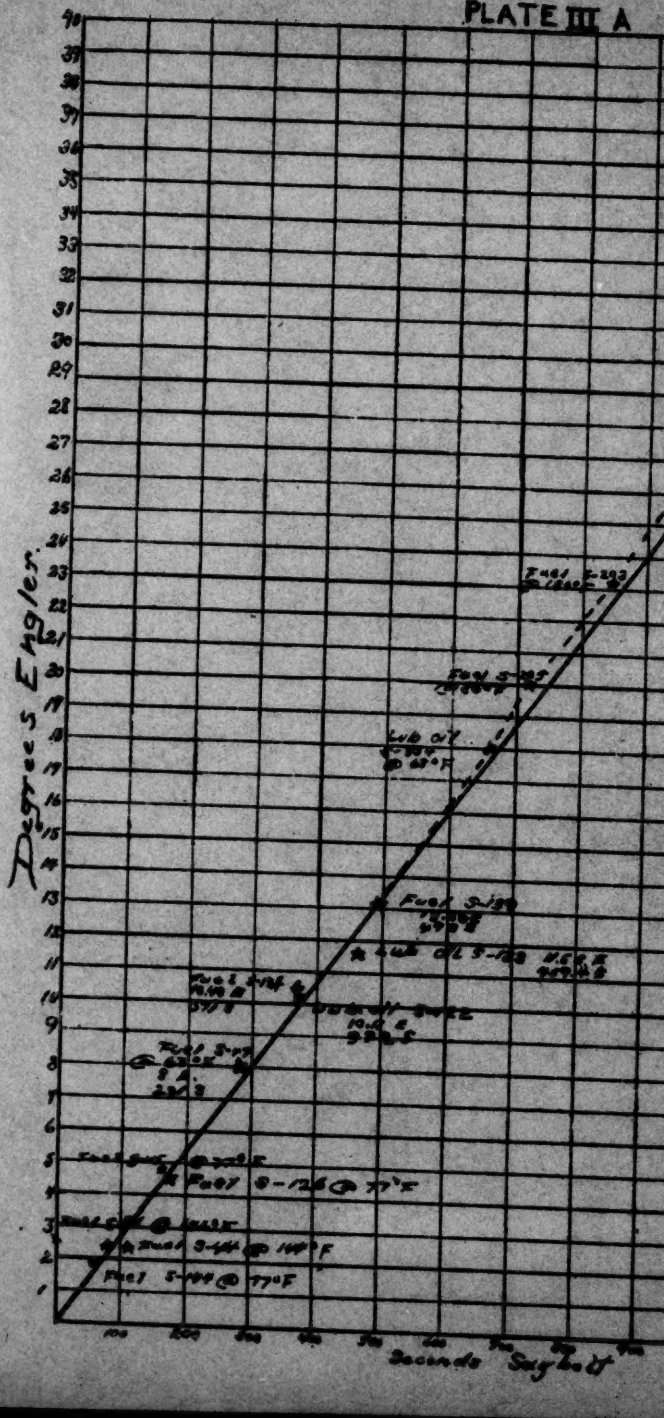
* Readings used in calculations.

85.90	85.90	85.90	85.90	85.90	
11.60	11.60	11.40	11.40	11.40	
.65	.65	.96	.96	.96	
.84	.84	.89	.89	.89	
1.61	1.61	1.45	1.45	1.45	
25.55°	25.55°	25.65°	25.65°	25.65°	
.9010	.9010	.9010	.9010	.9010	
195.	195.	215	215	215	
120.	120.	120.	120	120.	
3.06	3.06	3.17	3.17	3.17	
.3%	.3%	.2%	.2%	.2%	
19300.	19300.	19280.	19280.	19280.	
78.7	80.6	32.3	67.4	69.8	
98.60	98.29	99.86	——	——	
1.1222	1.1206	1.1041	1.1204	1.1167	
13.630	13.515	14.978	14.430	14.169	
15.296	15.145	16.537	16.167	15.881	
18.434	20.286	5.621	12.189	14.628	
20.687	22.752	6.208	13.697	13.364	
5.42%	5.61%	3.34%	——	——	
1.352	1.301	.375	.845	1.032	
13.623	15.120	3.780	8.509	10.399	
930.1	1032.3	709.8	710.0	710.0	
35.6	32.1	28.4	21.5	21.6	
69.6	70.8	63.9	64.1	53.5	
.001120	.001178	.001058	.000878	.000865	
.720	.752	.636	.619	.610	
.078289	.071888	.070387	.070991	.071161	
.078369	.071906	.071343	.071469	.071815	
169.4	178.2	196.6	194.7	194.8	
16.94	16.49	16.71	17.34	17.26	
17.04	16.58	16.80	17.41	17.40	
20.5	18.0	19.3	24.4	25.4	
——	——	——	——	——	
232.7	230.8	259.9	242.2	244.6	
4329	4765	5391	3468	3475	
28.9	19.5	21.5	25.9	26.9	

BTU	%	B.T.U.	%	B.T.U	%	B.T.U.	%	B.T.U.	%	B.T.U.	%
14643	76.91	14697	76.15	16049	83.84	15689	81.37	15302	79.78		
0	.00	60	.31	27	.14	21	.11	0	.00		
2469	13.00	2453	12.71	1304	7.81	2087	10.83	2222	11.53		
1378	7.26	1374	7.12	1229	6.42	1290	6.69	1306	6.77		
——	——	——	——	——	——	——	——	——	——		
76	.41	79	.41	42	.22	59	.30	41	.21		
433	2.28	637	3.30	430	2.25	194	.80	329	1.71		
19300.	100.00	19300	100.00	19280	100.00	19280	100.00	19280.	100.00		

76.91	76.15	83.84	81.37	79.78	

PLATE III A


Degrees Engler.
Seconds Saybolt
Fuel S-125 @ 77°F
Fuel S-144 @ 100°F
Fuel S-99 @ 77°F

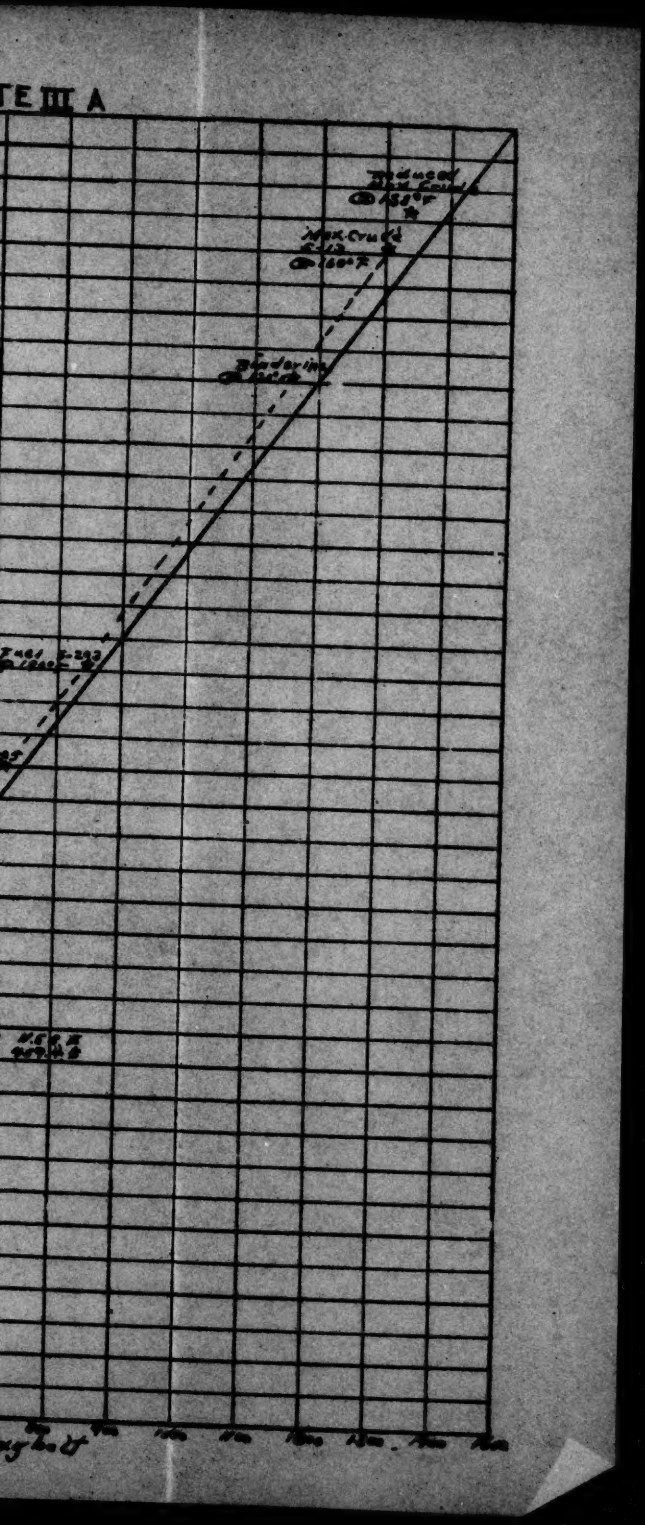
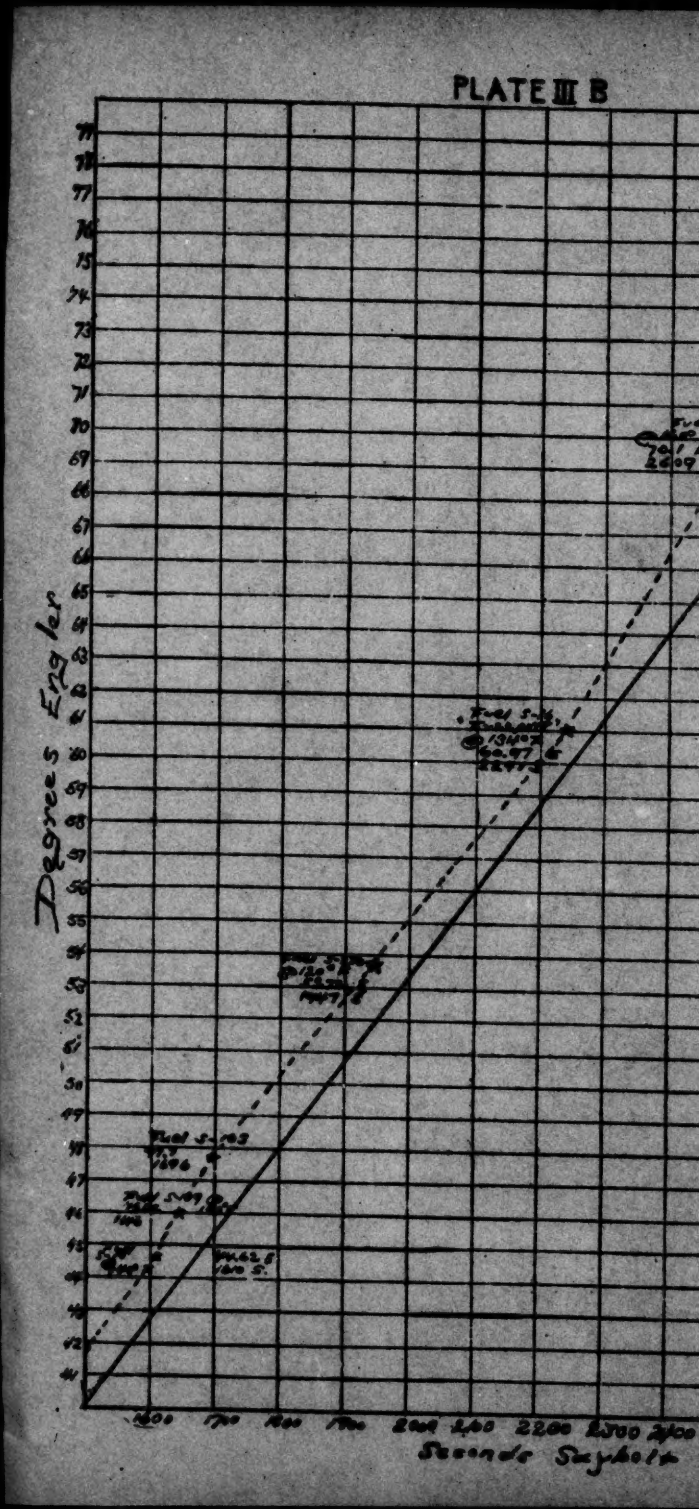
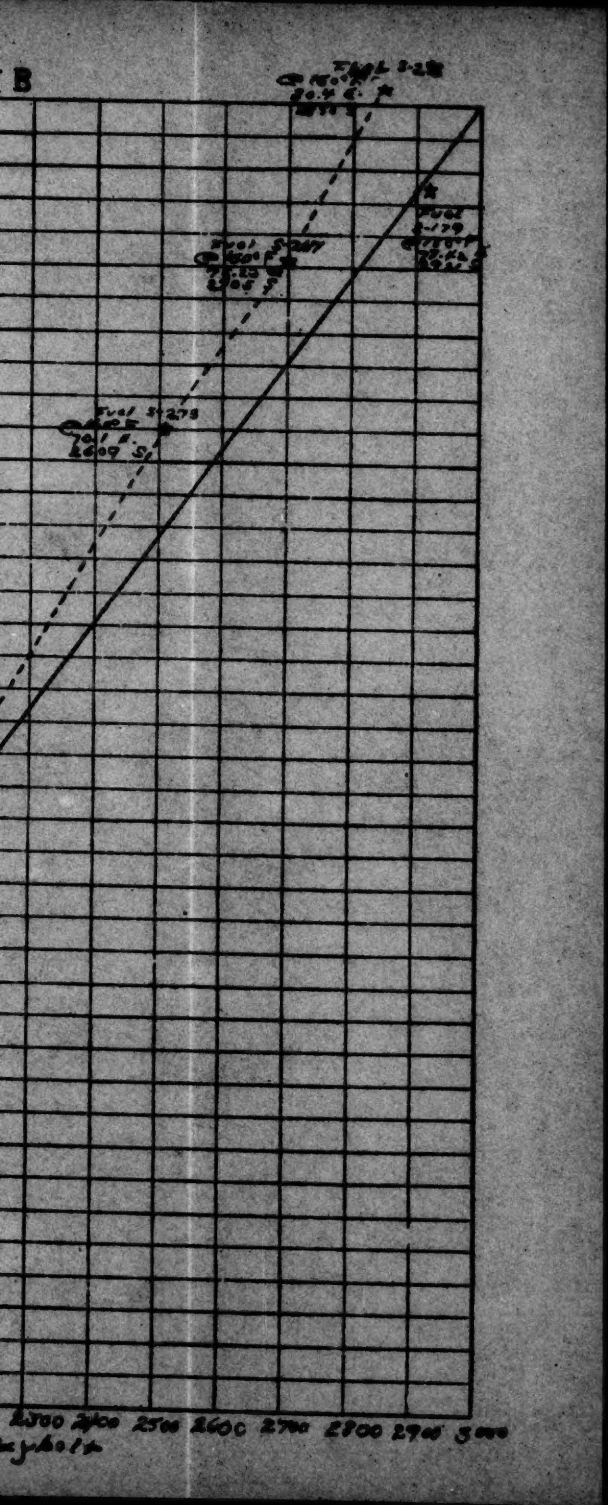


PLATE III B




B
2300 2400 2500 2600 2700 2800 2900 3000

several hours' steaming during which the individual burner capacity was pushed up to 2,000 pounds per hour. The record for the hour shows 2,238 pounds, while in the last half-hour the rate of combustion per burner reached 2,287, a safe margin over one ton of oil per burner per hour.

Owing to the three burners only being in operation in the large furnace it is to be noted that the oil burned per cubic foot of furnace volume was reduced to 8.9 pounds per hour.

I am again indebted to the Navy Department for permission to publish these remarkable tests (see Table III). The privilege is especially appreciated for the reason that the official report has not yet been completed.

These tests are so very recent that it is not yet possible to foretell their precise significance, but it is worthy of note that six months ago fifteen burners were being considered for each of the sixteen boilers of the battle cruisers now being laid down. On the basis of this last performance only five would be required, and seven would be a safe estimate. To develop 180,000 horsepower with 80 or even 112 oil burners is a very notable performance, but there can be no doubt about its being entirely possible with the equipment which has been perfected in this country. We shall no doubt hear the time-honored warning against putting too many eggs in one basket, but nevertheless I am convinced that the large unit, both in boilers and oil burners, has come to stay, particularly if it carries with it efficiency, accessibility, simplicity, safety and flexibility, which it most assuredly does.

I do not wish to be understood as saying that I think that oil burners atomizing a ton of oil per burner per hour will be immediately adopted for naval service, far less for the merchant marine, but the trend of modern development is certainly in the direction of larger units and the higher capacities made possible by the use of oil fuel.

Referring to the matter of flexibility, I think I am justified in discussing another recent development which I believe marks a distinct advance in the oil-burning art. It is well known that the mechanical atomizer of the usual design has a decided limitation in range. Unlike the steam atomizer, it cannot be operated at low capacity if it is designed for a high one. The lower limit with the oil pressures ordinarily used is approximately one-half the upper. The reason for this is very simple. The burner depends for its atomizing effect on centrifugal force induced by giving the oil a rapid whirling motion in the central chamber of the tip. This whirling motion, in turn, is produced by delivering the oil to the central chamber through eccentric passages, or small channels substantially tangential to the walls of the chamber. All the oil which enters the chamber through these channels finds an exit through an orifice concentric with the axis of rotation. Thus as the capacity of the burner is reduced by the obvious method of reducing the oil pressure, there is a lower velocity through the channels and friction losses become more apparent, until finally the spraying effect is reduced till the oil issues from the tip in a solid stream or in a spray too coarse and uncertain to give the desired results. In burners of the usual type this point, as stated, is reached at about one-half the maximum capacity of the burner.

In atomizers of the Thornycroft type, in which the size of the tangential channels may be reduced by an adjustment in the burner, the velocity through the channels is kept up to a large extent at low powers, and the range of these burners is much greater than in those of the usual type. But it has been found difficult in practice to make individual burner adjustment under service conditions, and this method has never become popular, even if it be assumed that a sufficiently wide range in capacity

is secured thereby, which is questionable. The only alternative, therefore, with burners of the ordinary design is to shut off some of the burners at low powers, or to change the size of tips, or to do both, which is not unusual. Both these methods are open to objection.

Another and radically new plan, however, has been devised which promises very satisfactory results. This consists of maintaining the whirling motion in the central chamber of the tip undiminished whatever may be the capacity desired of the burner; in other words, instead of delivering through the orifice all the oil which enters the burner chamber, as is invariably done in all other mechanical atomizers, a part of the oil supply is diverted or by-passed from the central chamber and returned to the pump suction, and the actual effective capacity or amount of oil which is sprayed into the furnace depends merely on the proportion of the oil that is by-passed to the total amount entering the burner. Thus, as the amount of oil entering the central chamber through the tangential slots remains at all times at the maximum, the whirling of the oil and the atomizing effect remains also at a maximum, so that a perfect spray is secured whatever the capacity, at low powers as well as at the highest rate. By this method the range of the atomizer is very greatly increased, and, instead of being limited at the low powers to 50 per cent of the maximum, a good spray can be secured all the way down to about 10 per cent of the total.

This method of spraying is the invention of Commander J. O. Fisher, U. S. Navy, now fleet engineer of the U. S. Atlantic fleet. He first applied it in an attempt to spray oil into the cylinders of an internal-combustion engine, where, owing to the intermittent action required, the ordinary mechanical atomizer was an entire failure. By introducing the by-pass a continuous flow was secured, even when the regular outlet orifice was closed. The application of the principle to atomizers for boiler work followed, with the result that a single burner without change of tip could be used over a greatly extended capacity range. The Navy Department has applied the Fisher burner with great success to small boilers of the launch type, where its wide range enables them to operate the boiler with only one burner. I have shown the navy design of this burner in Plate II. The company with which I am now connected has developed a modified form of the Fisher burner with which some interesting experiments have been carried out with the co-operation of the Navy Department. It is believed that the wide range in capacity will make the burner especially useful for fluctuating loads or in any installation where it is now necessary to change the tips or close off a portion of the burners.

It may be of interest to state that the burners having a maximum hourly output of 1,820 pounds which were tested at the Fuel Oil Testing Plant were successfully operated at 250 pounds, while smaller tips designed for a maximum of 800 pounds were operated at a minimum of 80 pounds per hour. In our design we have dispensed with the spindle shown in Plate II and we control the capacity entirely by an external valve in the by-pass return line, so that by the manipulation of this valve any number of burners can be simultaneously increased or decreased in capacity over the entire range. Also, the tangential channels and the outlet orifice are in a single piece.

A matter closely allied to the use of oil as fuel, in which there has been no advance, and if anything, a retrograde movement, is the measurement of the viscosity of oil. Unless I am grossly uninformed, we are no nearer a universal standard method of designating viscosity than ever, while, on

the contrary, not only are new viscometers and new scales being devised, but there is a tendency in certain quarters to use Engler "seconds" in the same way that Saybolt seconds are used, instead of dividing the time of outflow of the oil in the Engler viscometer by the time of outflow of an equal amount of water. The latter gives the true Engler scale, as we have been taught to regard it, and the use of Engler seconds only adds to the confusion.

It is probable that if the Saybolt viscometer had been obtainable by anybody outside the large oil companies ten or twelve years ago when the mechanical atomizer began to be used extensively, we should not now be using the Engler instrument at all in this country, notwithstanding its extensive use in Europe. But some method of determining viscosity became necessary and, as usual, the Germans were on the job and supplied us with Engler instruments. Various methods of interconversion of the two scales have been suggested, but the problem is not as easy as it looks, as the conversion factor is not constant throughout the viscosity range and it also varies with the character of oil. These differences are perhaps more academic than of real importance, as approximation to the viscosity results will usually serve.

Roughly speaking, the Engler degrees are 1/36 of the Saybolt seconds, *i. e.*, $S = 36E$, or $E = \frac{S}{36}$ where S = Saybolt seconds and E = Engler degrees (not seconds).

I am fortunate in being able to present a chart (Plate III) prepared by Mr. R. C. Brierly, chemist at the Fuel Oil Testing Plant, showing the relation of the two scales by actual test of samples of the same oil at the same temperature in the two types of viscometer. The solid line in the chart represents the ratio $E = \frac{S}{37.5}$ which is sometimes used for approximate determinations. The dotted line connecting the various points determined by test favors the ratio of 36 to 1.

Mr. Brierly states:—

"I am sending you enclosed the graphical representation of the theoretical Engler-Saybolt ratio and points plotted from our determinations. It is unfortunate that these ratio tests were not run at one or two certain temperatures, as temperature variation is undoubtedly a most important factor. However, I have noted oil temperature beside each plotted point.

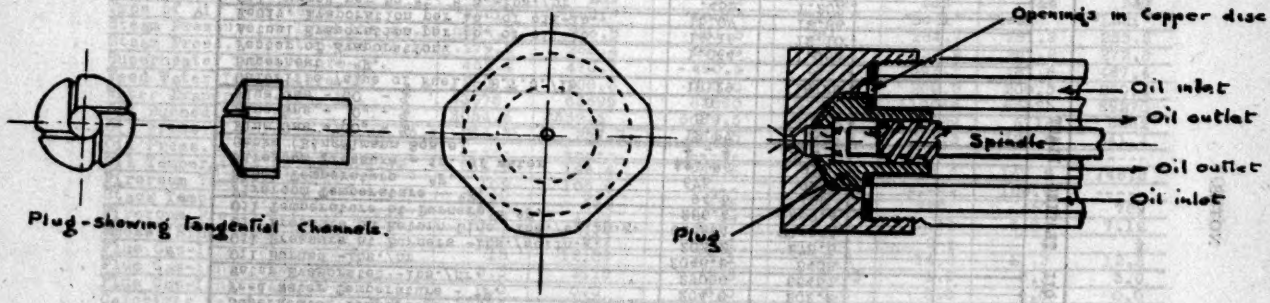
"As a matter of interest I ran a sample of 'Binderine'—paper pulp refuse—on both machines for comparison. I have shown it on the curve. Two of the most important references for the 1:37½ ratio are the U. S. Navy Oil Fuel Specifications of 1917 and H. G. Nevitt in 'Chemical and Metal Engineering' for June 23, 1920, page 1171."

It is to be hoped that some international standard method of measuring the viscosity of oil will be adopted, reconciling the Saybolt, Engler and Redwood scales and eliminating the periodic appearance of new proposals for determining viscosity. Only then shall we all be able to speak the same language.

Tabulation of Oils Used in Saybolt-Engler Comparisons.

Number.	Name of oil.	Sp. G. at 60° F.	Flash point °F.	B. T. U. lb. actual.	Remarks.
S-144	From U. S. S. Sub. K-1	.8581	185	*19588	Diesel oil.
S-44	J. H. Ref. Co., 3/14/18	.8940	185	*19322	
S-41	U. S. S. Mayrant	.8922	160	*19334	Used by Captain Bixby in trip to Norfolk.
S-126	U. S. S. S-2	.8930	225	19226	
S-45	U. T. L. X. 11555	.8892	180	*19355	
S-47	W. P. Boiler test, 4/2/18	.8917	200	19358	Mixed tanks.
S-122	Lub. oil G. Sub. U-154	.9301	350	.....	Grayish black loaded with animal oil.
S-124	Lub. oil G. Sub. U-164	.9294	360	.....	Muddy color smells like whiskey.
S-123	Lub. oil G. Sub. U-164	.9313	380	.....	Light oil.
S-139	W. F. test, 4/16/19, Tidewater	.9137	160	19102	
S-304	Vetol Motor oil	.....	405	.....	Light green.
S-205	Bunk "A" of Tex	.....	.....	.....	
S-203-4	Panuco, 1st car load	.9774	†120	18129	
S-13	Texaco Mex. Crude	.9760	86	18179	
S-11	Reduced Mex. Fuel	.9580	195	19033	
S-104	At. Ref. Bunker C	.9635	160	18621	
S-103	At. Ref. Bunker B	.9429	160	18994	
S-236	Oil used at Erie, Pa.	.9610	†185	18564	Jan. 6, 1920.
S-16	"Freeport"	.9700	142	18216	Mr. Pennycook, Freeport Mex. Fuel Corp., New Orleans, La.
S-278	T. C. X. 4832	.9516	†205	*18938	
S-267	Texaco 10.8° Be oil	.9945	†185	18682	
S-179	Schutte & Koerting exp. test oil.	.9877	‡	*18722	Sub. by Lieutenant Commander A. M. Penn.
S-273	Quimby test oil	.9974	180	*18664	Sub. by Wm. E. Quimby, Inc.

Flash point Pensky-Marten closed cup unless otherwise noted.
*Indicates given B. T. U. is calculated instead of actual.
†Indicates Cleveland open cup.
‡Unobtainable on account of expansion.



FISHER BURNER USED IN U. S. NAVY

COURTESY NAVY DEPARTMENT

PLATE II.

SUMMARY OF TESTS ON NORMAND BOILER.

PEABODY-FISHER BURNERS

Test Number				
Date	8-26-20	8-25-20		
Barometer-in. of mercury	- - -	- - -		
Duration of Test-hours	2	2		
Number of Burners	3	3		
Type of Burners	Peabody Fisher	Peabody Fisher		
Type of Air Registers.	Peabody	Peabody		
Steam Pressure on Drum -lbs. /sq.in.g.	240	240		
Steam Pressure below Stop -lbs. /sq.in.g.	- - -	- - -		
Superheater Temperature - °F.	- - -	- - -		
Feed Water Temperature - °F.	204.0	202.2		
Water Evaporated -lbs./hr.	29000	55467		
Oil Burned -lbs./hr	2048.5	5408		
Oil Pressure at Burners -lbs./sq.in.g.	203.2	273.6		
Oil Pressure on Return Line -lbs./sq.in.g.	Closed	Closed		
Oil Temperature at Burners - °F.	266.4	269.4		
Fireroom Temperature - °F.	94.5	105		
Stack Temperature - °F.	676	928		
Fireroom Pressure - in. of water	#1.86	0.8		
Smoke (Ringelmann Scale)	¼ - ½	¾ - 1		
Flue Gas - CO_2 - %	12.27	15.2		
Flue Gas - O_2 - %	5.1	0.5		
Flue Gas - CO - %	0	0		
Calorific Value of Fuel - B.T.U./lb	18129	18129		
Superheat - °F.	- - -	- - -		
Factor of Evaporation.	1.064	1.063		
Actual Evaporation per lb. of oil-lb	14.16	12.10		
Equiv. Evaporation per lb. of oil-lb.	15.07	12.86		
Oil Burned per Sq.Ft. H.S.-lbs./hr	.455	1.202		
Oil Burned per 00.FT.F.V.-lbs/hr	4.21	11.15		
Oil Burned per Burner per hr.-lbs.	682.8	1802.7		
Boiler and Furnace Efficiency - %	80.67	68.84		
Remarks.	# Draft in Uptake.			

Table II

FUEL OIL TESTING PLANT

NOTES.

TABLE III

SUMMARY OF TESTS ON WHITE-FORSTER BOILER

Test Number	44-3	44-6	44-11	44-7	44-12	*****	
Date	9-29-20	9-15-20	9-30-20	9-16-20	10-1-20	9-17-20	9-17-20
Barometer-in. of mercury	------	30.056	--------	29.94	------	30.10	30.10
Duration of Test-hours.	3	3	3	3	1	1	1
Number of Burners.	3	3	3	3	3	3	3
Type of Burners.	B&W	B&W	B&W	B&W	B&W	B&W	B&W
Type of Air Registers	"Lodi"	"Lodi"	"Lodi"	"Lodi"	"Lodi"	"Lodi"	"Lodi"
Steam Pressure on Drum-lbs.sq.in.g.	284.9	276.5	281.3	281.1	284.7	281.5	285.5
Steam Press.Superstop-lbs.sq.in.g.	279.3	268.4	269.9	261.4	255.9	267.2	272.5
Superheater Temperature-oF.	462.4	470.7	490.3	494.1	493.1	487.8	487.4
Feed Water Temperature-oF.	200.1	200.3	200.0	200.3	200.9	200.3	200.4
Water Evaporated-lbs. Hr.	45978	63695	83890	102222	119065	86625	89270
Oil Burned-lbs. Hr.	3051.4	4252.5	5089.5	7550.5	8989	6714	6862
Oil Press.at Burners-lbs.sq.in.g.	204.3	227.8	286.5	289.8	281.3	324.6	341.4
Oil Press.in Return Line	------	------	------	------	------	------	------
Oil Temperature at Burners-oF.	135.9	145.4	157.5	147.2	157.0	147.2	146.5
Firerom Temperature-oF.	101	105.3	------	102.3	------	100.9	------
Stack Temperature-oF.	567	633	675	714	780	667	709
Firerom Pressure-in.of Water	2.87	4.44	5.87	6.60	7.31	6.94	7.12
Smoke (Ringelmann Scale).	0-1	0-1	0-1	0-1	0-1	1	1
Flue Gas-Inner Pass-CO2-%	13	12.89	13.1	12.2	14.0	11.3	12.4
Flue Gas-Inner Pass-O2-%	3.3	3.15	2.8	3.65	1.9	5.3	3.0
Flue Gas-Inner Pass-CO-%	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Calorific Value of Fuel-B.t.u.	18939	18968	18994	18973	18994	18939	18938
Superheat-oF.	48.22	58.5	78.4	83.9	87.0	76.0	75.5
Factor of Evaporation	1.1016	1.1082	1.1257	1.1221	1.1239	1.1182	1.1173
Actual Evaporation per lb.of Oil	14.999	13.991	13.821	13.485	13.246	12.902	13.009
Equiv.Evaporation per lb.of Oil-lb.	16.525	15.506	15.559	15.182	14.887	14.427	14.535
Oil Burned per Sq.ft.H.S.-lbs.hr.	.404	.602	.802	1.002	1.1882	.888	.908
Oil Burned per Cu.ft.F.V.-lbs.hr.	4.057	6.06	8.082	10.09	11.969	8.940	9.138
Oil Burned per Burner per Hr.-lbs.	1009.1	1517.5	1213.9	1515.1	1498.4	2238	2287.3
Boiler and Furnace Efficiency-%	81.29	79.24	78.98	77.30	75.05	73.92	74.47

Remarks

*****The tests of 9-17-20 are unofficial because of three tubes in boiler being plugged

NOTES.

THE EFFECTS OF HOLES, CRACKS AND OTHER DISCONTINUITIES IN SHIPS' PLATING.*

By Professor E. G. Coker, D. SC., F.R.S., and A. L. Kimball, M.E.E.

The present paper is an extension of the subject matter of two former papers† in the Transactions of the Institution of Naval Architects, in which the effects of various discontinuities were examined to find the alteration in the stress distribution produced thereby, principally in tension members. It confirms, in general, some of the principal results of the mathematical theory of the stress around an elliptical hole in a wide plate, which formed the subject matter of a paper‡ by Professor C. E. Inglis contributed in 1913. Viewing a crack as the limiting case of an ellipse, in which the minor axis is vanishingly small, it is easy to show, both from calculation and experiment, that a crack inclined to the direction of stress produces a great concentration, which approaches an infinite value. A practical way of reducing the stress intensity is to drill out the ends of the crack, and the experiments described here show what effect this produces, and also the change in stress produced, by forming elliptical cavities at the ends instead of circular holes.

At the date of the earliest of the papers mentioned above the stress which occurs in deck plating at a hatchway was also examined, but only for the purpose of determining the stress across the weakened principal section, whereas the principal point of interest arises at the change of section of which some measurements are now contributed, to show how great is the stress experienced at the joint of the diminished section to the main body of the member.

The experimental methods used have been sufficiently described before, and it is, therefore, only necessary to recall that the complete determination of stress in a plate subjected to loading in its own plane can be completely solved experimentally without reference to any theory of stress distribution, provided it may be assumed that the plate is not so thick as to cause a variation of stress across the section, and for applications to other materials that load systems applied to each edge boundary are in equilibrium separately, as is the case in all the examples described here.

If these simple conditions are fulfilled, the solution of any stress problem at a point in a plate, whatever may be its form, can be determined by three sets of measurements. There are:

(1) The directions of the principal stresses P and Q.
(2) The value $(P - Q)$ from an optical measurement.
(3) The value $(P + Q)$ by a mechanical measurement.

It is possible, however, to dispense with the third measurement in a complete solution, but it is generally more convenient to obtain this. In many practical problems, however, one or more of these measurements can be dispensed with. For example, at a point along a boundary where there is no applied load, the values of $P \pm Q$ reduce to either P or Q, since the other stress must be zero, while the direction of the existing stress is along the boundary.

This latter case gives an important simplification, since in many instances the only important stress is at a boundary.

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* Paper read before the Institution of Naval Architects, March 25, 1920.

† E. G. Coker: "The Determination, by Photo-Elastic Methods, of the Distribution of Stress in Plates of Variable Section, with Some Application to Ships' Plating," Transactions of the Institution of Naval Architects, 1911. E. G. Coker and W. A. Scoble: "The Distribution of Stress Due to a Rivet in a Plate," Transactions of the Institution of Naval Architects, 1913.

‡ C. E. Inglis: "Stresses in a Plate Due to the Presence of Cracks and Sharp Corners," Transactions of the Institution of Naval Architects, 1913.

An interesting example is presented by the case of an elliptical hole in a plate in which the ratio of the length ($2a$) of the major axis to the length ($2b$) of the minor axis is varied between equality and infinity giving a circle for one limit and a fine crack at the other.

It is not proposed to consider circular holes here, except incidentally, as these have already been examined with some care. The case of the elliptical hole has not been examined experimentally so far as we are aware, although an exact solution is known for such a hole in an infinitely wide plate. In order to verify this theory, various elliptical holes were cut in a plate of clear nitrocellulose 5 inches wide and 0.155 inch thick, to which a uniform tension load was applied in a testing machine of three tons capacity. The arrangement for loading the test piece is of the simplest kind, Fig. 1, and consists of a pair of plates A secured to each end by a

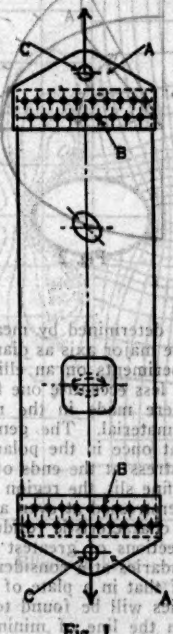

Fig. 1

number of small pins B to distribute the pull applied by the machine through centrally placed pins C. The positions of the holes cut in this plate are indicated approximately in this figure.

In such an arrangement nearly the whole length of the uncut plate should be in pure tension, and this was found to be so by optical measurements, except for a short distance from the ends.

It was considered sufficient to examine three cases of elliptical holes, and these were selected somewhat arbitrarily by taking an ellipse with a major axis 1.125 inches long perpendicular to the line of pull, and having a minor axis 0.375 inch. This hole was subsequently altered to one with major axis 1.2 inches, and the minor axis was increased to 0.8 inch. In a third example this latter form of hole was intended to have the major axis at 45 degrees to the line of pull, but through an error in cutting out this line was found to have an inclination of 49 degrees to the line of pull, and was therefore somewhat inconvenient for calculation purposes. For convenience of reference in all these cases, any point A, Fig. 2, on the

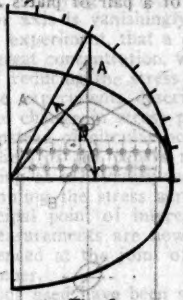

Fig. 2

boundary of these holes is determined by means of the eccentric angle β for a circle described on the major axis as diameter. It will be convenient to describe the earlier experiments on an elliptical hole athwart the line of pull by reference to the less eccentric one first, although, as mentioned above, the experiments were made in the reverse order to economize skilled workmanship and material. The general features of the stress distribution are apparent at once in the polariscope, the effect of a pull being to cause an intense stress at the ends of the major axis, and if the elliptical hole reduces to a fine slit, the region around the ends of this latter becomes very much overstressed at even a small load, and shows this on removal of the load by characteristic residual stress effects. It is also to be noticed that the directions of greatest stress intensity away from the hole approach the boundaries at a considerable angle to the axial line, and this leads to the belief that in a plate of narrow width the stress at the outer straight boundaries will be found to reach a maximum at two places equally distant from the line of minimum cross-section. An examination of a plate with an elliptical hole in it of considerable size shows this to be the case and confirms some recent measurements of the effects of moderate-sized circular holes in tension members.* Apparently the stress at the minimum section of this latter member has, in general, maximum values at the inner boundary and minimum values at the outer one.

* E. G. Coker, K. C. Chakko and Y. Satake: "Photo-Elastic and Strain Measurements of the Effects of Circular Holes on the Distribution of Stress in Tension Members," "Transactions of the Institution of Engineers and Shipbuilders in Scotland," November, 1919.

In the elliptical hole the same effect is noticed, and is probably connected with the fact that in such a case the lines of principal stress, having a general direction parallel to the line of pull well away from the hole, curve first one way and then the other as they get farther from the hole, so that there must be a change in stress at the boundary if this latter is only a finite distance away. This effect is noticeable even in the present

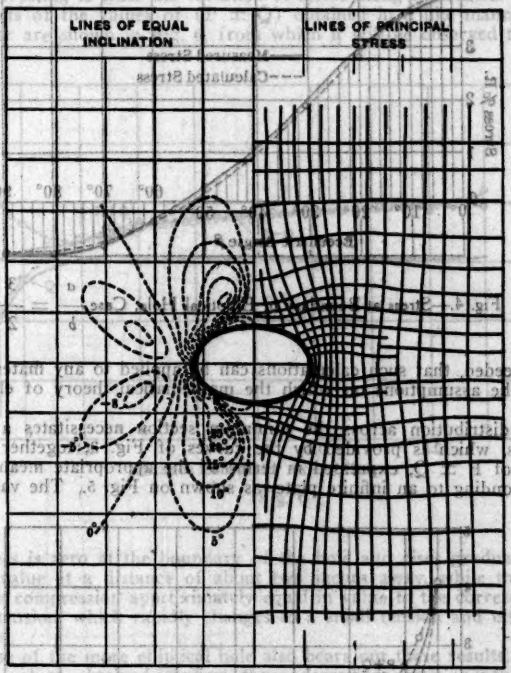

Fig. 3.—Elliptical Hole $\frac{b}{a} = \frac{2}{3}$

case, and the lines of principal stress, determined experimentally and shown in Fig. 3, exhibit this clearly. The lines of equal inclination, from which these curves are derived, are also marked on the left-hand side of the diagram. An examination of the stress at the elliptical boundary shows that its variation is very great. Commencing at the extreme end of the major axis, where the tensile stress is most intense, Fig. 4, it gradually diminishes to zero at a point having an eccentric angle of 54 degrees, and becomes a compression stress which reaches a maximum at the end of the minor axis.

A comparison of these observed stresses with the calculated values obtained from the appropriate equations of Professor Inglis' paper shows a gratifying agreement, as the figure shows, and affords a proof, if any

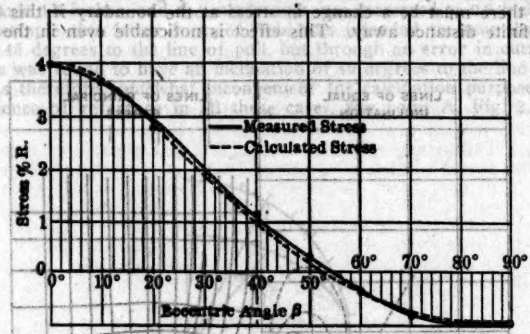

Fig. 4.—Stress at Boundary of Elliptical Hole, Case $\frac{a}{b} = \frac{3}{2}$

were needed, that such calculations can be applied to any material which obeys the assumptions on which the mathematical theory of elasticity is based.

The distribution across the minimum section necessitates a complete analysis, which is provided by the curves of Fig. 3, together with the values of $P \pm Q$, expressed in terms of the appropriate mean stress R corresponding to an infinite plate, as shown on Fig. 5. The values of P

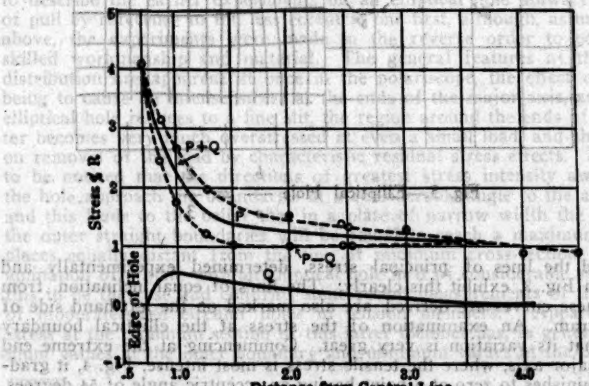

Fig. 5

and Q have been separated from these observations, and are shown by firm lines. It will also be noted that in addition to the great intensity of the tension at the inner point of the boundary there is a perceptible tensional cross stress near the edge, and as the ellipse tends more and more towards a fine horizontal crack, the maximum value of Q increases and also approaches the boundary, although it can never actually reach it in any practical example.

It is interesting to trace the variation of stress along the central axis by observations of the values of $(P \pm Q)$ obtained in a like manner, and these latter are shown in Fig. 6, from which it will be observed that the

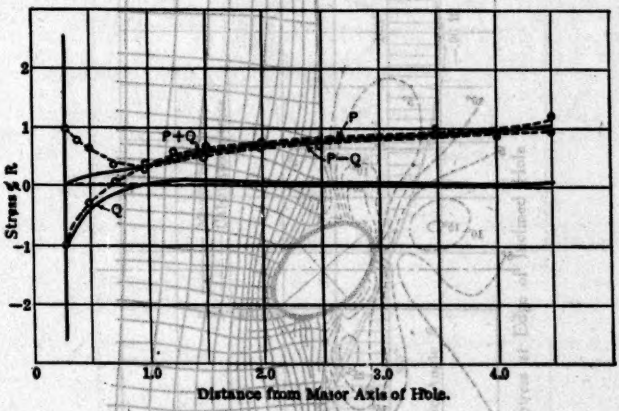

Fig. 6

axial stress is zero at the boundary of the hole and rises gradually to a constant value at a distance of about two inches away, while the cross stress is a compression approximately equal in value to the corresponding uniform tension, which rapidly changes to a small tension and ultimately disappears.

The case of the more elliptical hole also bears out these results, and, in fact, the values obtained confirm those described above, so that it is hardly worth referring to them in detail, except to point out that all the experiments for this type of hole agree in showing that the maximum stress is always approximately $(1 + 2\frac{a}{b})$ times the stress corresponding to that of the infinitely wide plate, where a and b are the appropriate values of the axes, and the hole is small.

The case of the elliptical hole inclined to the line of pull presents some interesting features not shown by the former cases. In the first place the maximum stress is not at the ends of either major axis, but is found at or near the point where the tangent line parallel to the line of pull touches the ellipse. The lines of stress, as may be surmised, only possess a bi-lateral symmetry, as is shown by the experimental determination, Fig. 7, in which one also observes the tendency of the outer lines to bend out-

wards at some distance above and below the hole, and not in quite the same way above and below the transverse central plane. They appear, in fact, to indicate maximum stresses at the outer straight sides at unequal distances from the central cross-section, and polariscope observation confirms this.

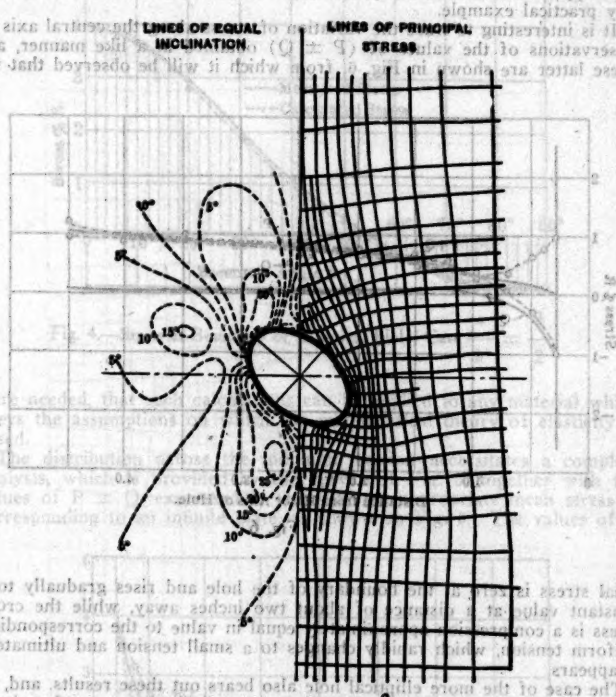

Fig. 7.—Elliptical Hole $\frac{a}{b} = \frac{3}{2}$, Inclination 49 Degrees to the Line of Pull

The stress distribution round the hole has been determined and compared with the theoretical expression of Inglis, and a very fair agreement is found, as Fig. 9 shows, in which the two are compared by plotting the stresses observed and calculated on a base derived from the eccentric angle β. The stress is found to vary between 3.43 R and —0.96 R, and is in good agreement with the theoretical expressions. It is also of interest to note that when the stress distribution is examined across the plane perpendicular to the line of pull and passing through the center of the ellipse, it is found to account for the total pull within one per cent.

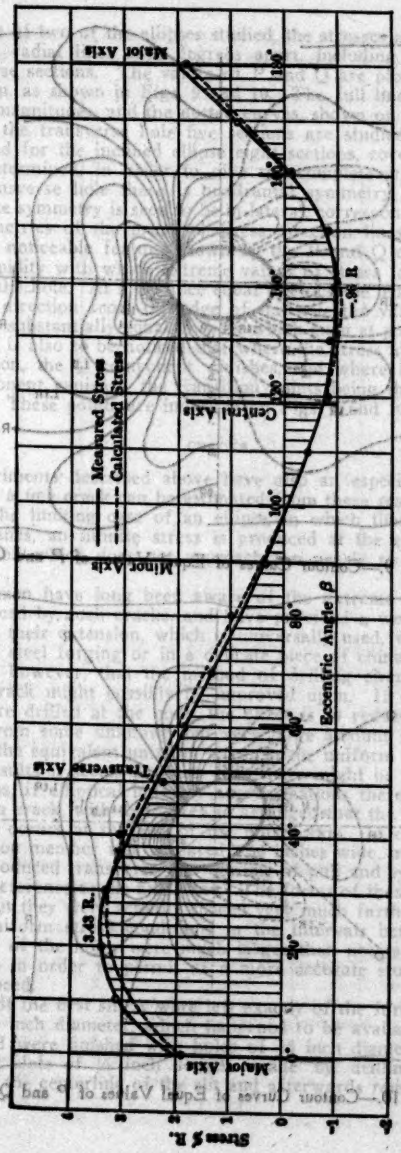

Fig. 8.—Stress at Edge of Inclined Hole

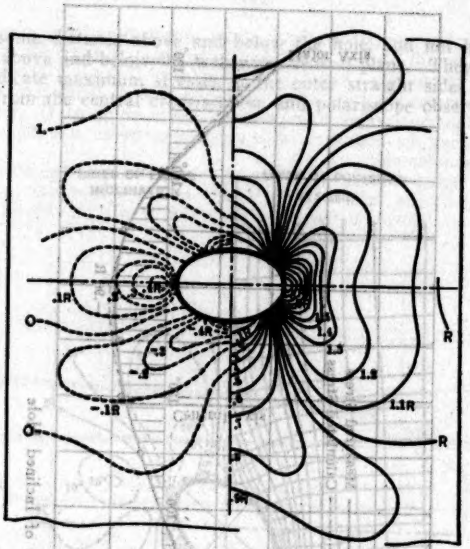

Fig. 9.—Contour Curves of Equal Values of P and Q

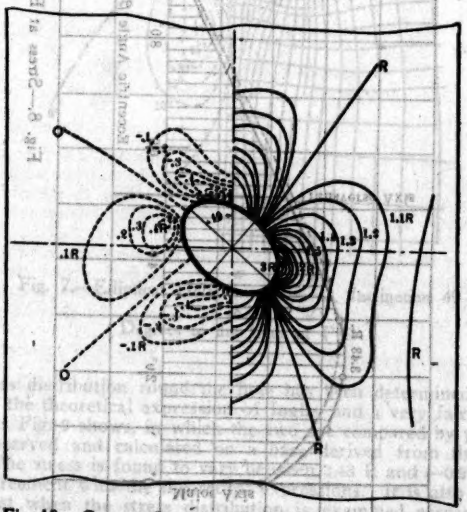

Fig. 10.—Contour Curves of Equal Values of P and Q

In the case of two of the ellipses studied, the stresses are determined at points along radial lines 22½ degrees apart, including the longitudinal and transverse sections. The values of P and Q are plotted and contour curves drawn, as shown in Figs. 9 and 10. The full line contours correspond to P magnitudes, and the dotted curves, shown on the left, to those of Q. For the transverse hole five sections are studied, to include one quadrant, and for the inclined ellipse eight sections, covering half of the area, are determined, in order to give the complete stress distribution. For the transverse hole there is quadrantal symmetry, and for the inclined one the symmetry is seen to be bi-lateral, corresponding in each case to the symmetries of the principal stress direction lines shown in Figs. 3 and 7. A noticeable feature shown by the P and Q contour curves is the great rapidity with which extreme values of stress fall off on leaving the edge of the hole. At a distance equal to twice the length of the major axis in any direction from the edge of the hole the values of P and Q have become substantially equal to those which exist at points remote from the hole. It is also to be noticed that where the stress at the edge of the hole is tension, the Q component vanishes, and where it is compression the P component vanishes, the transition points being the four points of zero stress. These points are indicated on Figs. 9 and 10.

CRACKS.

The experiments described above have also an especial interest, since the effect of a fine crack can be estimated from these results. If we consider it as the limiting case of an ellipse, in which the minor axis ultimately vanishes, an infinite stress is produced at the extreme ends for any inclination which does not approach too nearly to the direction of the pull.

Practical men have long been aware of the extreme concentration of stress produced by such cracks, and have provided a well-known remedy for stopping their extension, which is universally used, whether it be for a crack in a steel forging or in a delicate piece of china. It occurred to the authors, however, that the method of drilling circular holes at the ends of a crack might possibly be improved upon. If circular holes of small size are drilled at the ends, the effect is to reduce the stress concentration from some unknown but very large amount to approximately three times the equivalent uniform stress in the uniform plate. From the preceding results it seems probable that there might be a further reduction of stress, if elliptical holes or approximations thereto are drilled at the ends of a crack, with their axes so arranged that the position of maximum stress occurs at the end of the minor axis. In order to test this view, a tension member was prepared 1.56 inches wide, in which four fine slits were produced transversely to the line of pull and symmetrically disposed with reference to the axial line. The forms of these slits are shown in Fig. 11, but they were actually spaced very much further apart in order to obtain uniform stress conditions in the intervals between them, and the contours of the holes were much larger than necessary for practical requirements in order to permit of a more accurate study of the stress effects produced.

The ends of the first slit A were left exactly of the form produced by a drill of 1/32 inch diameter, which happened to be available; the ends of the second B were finished with holes of ¼ inch diameter; the third C with vertical slots of ¼ inch length, made by drilling ⅜-inch holes tangential to the centerline of the slit and afterwards removing the angles

left by the drill. The fourth D was produced in a similar fashion, and the outer part curved to a radius of 0.22 inch to give an approximately elliptical contour.

The total width of the slit was ½ inch in all cases, except for D, and an allowance for the extra stress produced by the increase in the size of the

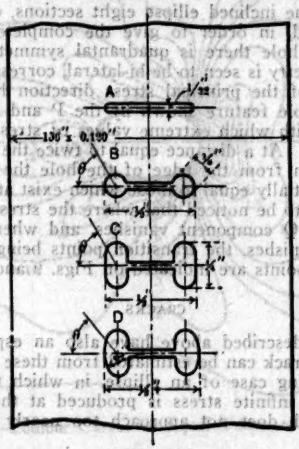

Fig. 11

latter is, therefore, made for comparison purposes. The specimen so prepared was then subjected to a load of 100 pounds (corresponding to a mean average stress of 790 pounds per square inch) in a small testing machine and the stresses round the contours determined for a sufficient number of points to make a comparison effective. The values so obtained are shown in Fig. 12, in which all are compared by aid of angles θ as abscissæ

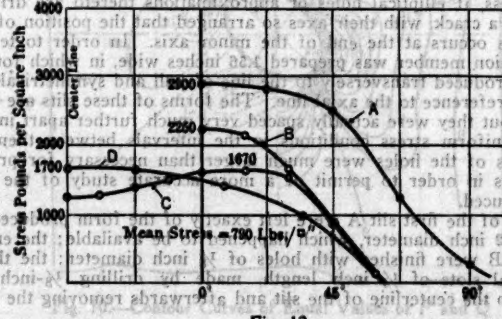

Fig. 12

and stress as ordinates, so far as this is possible, and the remaining parts of the contours of C and D by linear measurements along the contour. The distances in the two latter cases are not quite the same, but the difference is not material. The comparison at once shows that for the simple case taken it is possible to materially reduce the high stress caused by a semicircular ending to a crack by drilling out a nearly complete circle, and proves the value of the practical means usually adopted, especially for a hair-line crack, where presumably the stress effect is very great. The substitution of form C effects a still further improvement by lowering the stress intensity from 2,250 to approximately 1,670 pounds per square inch, with a minimum value at an angle of $\theta = 15$ degrees approximately, and a minimum value at the central section of about 1,250 pounds per square inch. The form D, as might be expected, gave a maximum stress at the central line, and, as measured, corresponded to a stress of 1,700 pounds per square inch when the extra width was taken into account. Practically there seems little to choose between these two last forms, although the experiment is not decisive one way or the other. Having regard to the measurements on elliptical holes it is possible that form D is actually the best one to adopt if the proper shape of elliptical hole is chosen.

STRESSES AROUND HATCHWAY OPENINGS.

It is unnecessary here to refer particularly to the case of a large circular hole in a tension member of finite width, as these have been examined in an earlier paper, but there are many other types of discontinuity which are of importance, and an interesting practical case to which shipbuilders have devoted considerable attention is that of hatchways in ships' decks. They are probably subjected to complicated load systems not easy to define or to produce in the laboratory, but a simple type of loading, such as that of a pull, serves at any rate to bring out some characteristic features. The chief places of interest in all such cases of sudden discontinuity appear, in general, to be at or near the sudden change of section, and the case selected shows this quite well, although it would probably be easy to obtain a more characteristic one.

This example consists simply of a square hole of 2-inch sides cut in a plate of 5-inch width and having rounded corners of half an inch radius, the sides being parallel and perpendicular to the pull.

The large radii at the angles were chosen with the object of getting precise measurements at points of this part of the contour, and are greater than is usual in practice. The maximum stress occurs at the interior contour and near the joint of the rounded corners, with the sides parallel to the line of pull. There are also indications of a subsidiary maximum along the external contours at a considerable distance from the hole. These observations are amply confirmed by the measurements shown on Fig. 13, in which the stress distributions around both contours are shown for a total load of 390 pounds, giving a mean stress of 840 pounds per square inch at the reduced central section. In addition, the stress distribution across this latter section is shown by aid of observations of $(P \pm Q)$ shown in dotted lines, from which it may be observed that the cross-stress Q is negligibly small, while the maximum value of P at the interior contour along this section is 1,290 pounds per square inch.

Integrating the curve there shown and comparing it with the pull applied, it is found that the mean error of observation is approximately 3 per cent, which is a useful test as giving an idea of the general degree of accuracy obtained. The actual maximum is at the commencement of the angle and just beyond the joint of the straight and curved portions, with

a value of 1,730 pounds per square inch, or 2.06 times the value of the mean stress at the central cross-section. It is also easy to show that as this curve lessens in radius the stress increases and becomes dangerously great, even with a moderate load.

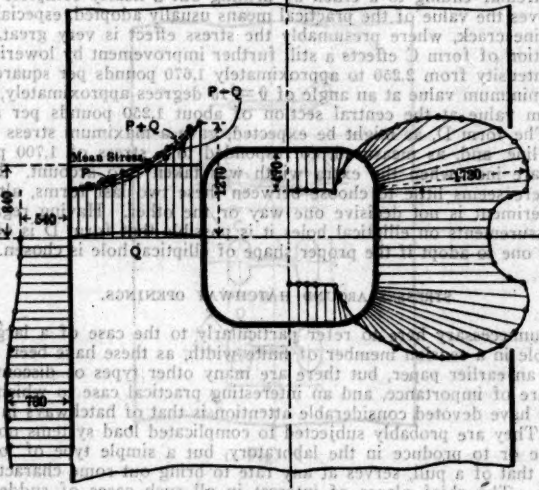

Fig. 13

As we pass still farther along the curve, the stress rapidly diminishes and finally changes sign and becomes a compression of fairly uniform value along the plane contours. The outer contours also show the characteristic feature, already mentioned above, for another case of a minimum value at the central section with maxima on either side, but in this case these latter values are not very pronounced owing to the distance of the hole from the outer boundaries.

It is probable that a study of other cases of hatchway openings of more immediate practical importance than the one described here would afford information of value in practical construction.—"Marine Engineering," Nov., 1920.

THE NAVAL SITUATION.

The Government have acted wisely in referring the present naval situation to the Committee of Defense, as the problems which have arisen as a consequence of war experience are of the first magnitude, and can only be solved by the closest investigation alike from the strategical and tactical standpoints and from the constructional point of view on the fullest scientific basis. It was natural that controversy should arise not only as regards the will of the Government to maintain our naval supremacy, but also as to the most suitable type of ship calculated to secure this end. That the United States is building sixteen capital ships and Japan eight of

designs which must more or less embody the lessons discovered in the war, while we have not a single new ship on hand, capital or otherwise, gives, as the French say, "furiously to think."

Controversy, however, whether on the platform, in the press or in Parliament, seldom produces satisfactory results, except by acting as a stimulant to thought, because those engaged are often too prone to make points against those who do not see eye to eye with them rather than to elucidate with conscientious earnestness of purpose the problems under review. It can scarcely be said that the discussion which has been going on recently with regard to the present naval situation is free from this serious blemish, so that the removal of consideration to the calmer and more judicial atmosphere of the Committee of Defence is acceptable. This is especially the case as all able to contribute towards a solution will under these circumstances have the privilege of expounding their views before the committee, with full confidence that due weight will be given to them.

There has been some measure of confusion of thought as to whether the Committee of Defence is the most appropriate tribunal before which the issue should be brought. It has been contended, for instance, that a committee composed of naval officers, shipbuilders, engineers and scientists corresponding to that appointed by Lord Fisher to evolve the elements of the Dreadnought design, would have been more appropriate. But in that instance there was no uncertainty as to the type of fighting ships aimed at. Lord Fisher's desire was to produce "an all big gun ship" with the other requisites to efficiency based on the most satisfactory compromise, and obviously this involved constructional questions of great importance. Today the primary problem is one which in the first instance, at all events, is concerned with the type of ship. Once more the question is surface ship or submarine, perhaps even restricted to battleship or submarine.

Years ago the French, who had done more in the initial stages of submarine boat design, were faced with this problem, and for a time at least they were certainly disposed to place greater reliance on the submarine than on the battleship. It is no disparagement to our present relationship with our allies today to recall that their submarine programme was then, in the early years of the century, regarded as a menace to the British merchant service. Indeed, Commander Sir Trevor Dawson then foreshadowed what became a reality in 1917 by picturing the tremendous hazard of a fleet of submarines along the highways of commerce into and out of British ports. [Parenthetically it may be asked if we ought not to have been better prepared with weapons against submarines.] The French ultimately returned to adherence to surface ships, including capital ships, while maintaining the building of submarine craft.

It is true also that prior to the war there were strong advocates amongst experienced British naval officers for the supercession of battleships and battle cruisers by submarines, and many predictions were made as to the impotence of these capital ships in the event of war with a great European naval power. None of these predictions was fulfilled. We were told that we would require to bottle up our great battle squadrons in harbor. It is even said now that such fleets while in Scapa Flow ought to have been sunk by the German submarines. No attempt was made to do this, although the advocates of the submarine admit that the conditions were admirably suitable for such a great achievement.

Why was this prediction not fulfilled? The Committee of Defence will be expected to consider this. Far from the fighting fleets of Britain being confined to their harbors, Lord Jellicoe, in his flagship, led his fleet in the first sixteen months of the war alone on cruises seeking for the enemy

fleet over a distance of 44,680 miles, and Lord Beatty in subsequent cruises covered equal distances, and yet not a single battleship was sunk by submarines. It is true that several of the older ships were so disposed of early in the war by the Germans, but this was largely due to their lack of speed, so that the question of design and construction of surface ships to combat torpedo attack from submarines becomes one of the problems to be tackled by the Committee of Defence in their investigation, and to this we shall refer presently.

Again, at the Battle of Jutland, no German submarine was engaged, and from German accounts we learn that over 100 torpedoes were fired by surface ships, but the only battleship hit was the *Marlborough*, and by reason primarily of the admirable system of "bulge" protection she was able to return safely to the base. It can scarcely be said that the Battle of Jutland was fought to a finish owing to conditions of invisibility intervening, or more might have been learned as to the power or impotence of the battleship. At the same time the fact remains that as a result of the battle the German fleet never again left its base.

A contention of the advocates of abolishing the battleship is that a main function of the navy is the protection of our overseas commerce and no one will controvert this view. This involves the destruction of the enemy fleet, however it may be composed. The *Emden* had more British merchant ships to her credit than any one submarine; the same can be said of some of the specially-adapted merchant ships of the German fleet. Without surface ships these marauding cruisers could not have been combatted. It is difficult, too, to reconcile the supercession of surface ships with the triumph at the Falkland Islands, which had immense influence on the issue of the war and on the safety of merchant shipping on distant seas.

There is, however, much to be said in favor of the submarine because of its success in torpedoing merchant ships in the narrow seas. Certainly the results during the period when there was unrestricted submarine attack between April and June of 1917, had a most disturbing effect upon the *morale* of the British people. In these three months 377 merchant ships of a gross tonnage of about 1,228,000 tons were sunk with a loss of life of nearly 2,000. This gave some credence to the view that the German submarine policy might ultimately succeed. But this promise was short-lived. On the one hand it may be that Germany could not maintain the submarine fleet at adequate strength. We now know that Germany began the war with 28 submarines, the two largest being of 675 tons with six torpedoes, that she completed three before the end of 1914; that in 1915 sixty-two were delivered, in 1916 ninety-five, the size having increased to 860 tons; in 1917 one hundred and thirty, the largest being 1,500 tons, and in 1918, up to the armistice, eighty-one, the largest being 2,158 tons. Thus they brought into action from first to last 399 submarines. There were sunk by British warships and auxiliaries 207 German submarines, and it is interesting to note that the number of losses in each succeeding year was progressive: Thus for the first five months of the war in 1914 the losses were five, in 1915 twenty, in 1916 thirty, in 1917 seventy-two, and in 1918, up to the time of the armistice, eighty. The increase in the destruction of German submarines beginning in 1917 was due to the splendid work done by scientists in devising means to combat the unrestricted attack by the Germans on our merchant fleet.

It is not necessary here to enter at large upon the work thus done for the protection of merchant ships, in the construction of special types of attacking craft, in the discovery of hydrophones for the location of submarines and in the design of depth charges and other weapons for de-

stroying them. The influence of such means and the possibility of improving and augmenting them are subject-matters for the consideration of the Committee of Defence. The effect of these instruments of destruction is found in the fact that the losses of British merchant ships due to submarine attack greatly diminished, the numbers per month in the summer of 1918 being reduced to about 40 ships, as compared with 125 per month in the period of most intense submarine attack. In part this was due no doubt to the effect on the *morale* of the crews of German submarines, an effect which largely led to the revolution amongst the *personnel* of the German Navy. This may not be a factor which should influence the decision of the Committee of Defence, but serious consideration must be given to the potentialities of counter-attack on submarines.

On the other hand, equal care must be given to what may almost be regarded as inherent difficulties in the design of submarine boats. The Germans at the time of the armistice had ordered 400 new submarines—in itself a proof that they had still confidence in this style of craft—but an examination of the boats surrendered by the enemy to Britain showed that little progress had been made towards advance in design and particularly in propelling machinery. It is to the credit of the British service and engineers that we had the fastest submarines on the surface as well as submerged. It is not possible to combine these advantages in one craft and, moreover, the fast surface vessels of the "K" class had an exceptional combination of machinery—geared turbines, electric generators and motors, and oil engines—and consequently great weight resulted. Even in the vessels with internal-combustion engines and electric machinery the weight was unavoidably high, and consideration will require to be given to the problem as to whether it is possible to achieve better results in weight per unit of power developed. It is gratifying to know that the engineering laboratory of the Admiralty has been extended, and any expenditure in this direction will be most profitably incurred. The four-cycle single-acting engine is the most largely favored, but the desire of the engineer is for a satisfactory workable type of two-cycle double-acting engine, and it will be well if the attention of the engineering laboratory of the Navy is directed towards the production of such a prime mover not only for submarines but for other types of craft. The Admiralty can be assured of the encouragement and assistance of private firms who have done much in this country for the development of the internal-combustion engine. Another inherent difficulty of the submarine is that increased size involves greater time in diving, which is the preferable system of submergence when the vessel is under way; and in action the chances are that this will be the prevailing procedure. As the submarine grows larger, too, the effect of shallow water becomes more and more serious, so that before reliance can be placed on the submarine as the sole or principal unit of a navy for attacking or defending commerce many scientific problems must be solved.

With the capital ship, on the other hand, improvements can be effected in practically every element in design. It is no stretch of imagination to assume that since the war those responsible for design at the Admiralty have been giving full and continuous consideration to all these problems. During the war, amazing developments were made and the skill and experience of the naval architects, engineers, ordnance and torpedo officers were proved by the successes recorded.

We have referred to the improved protection against torpedo attack, and it need only be mentioned also that the dropping of bombs by aircraft was combatted by increased horizontal protection. Thus even the new arm of the service may be checkmated as far as the capital ship is

concerned, but it will be a powerful factor against the submarine. The whole question of the possibilities of aircraft will have to be considered from the strategic and tactical points of view, and it would seem as if a serious error had been made from the administrative point of view in allowing the new arm of the service to pass absolutely from the Admiralty control. It is of the utmost importance that research work should continue in connection with the constructional features of all types of aircraft.

All this may seem to justify the view that a special committee should have been appointed to consider the question rather than that it should have been left to the Committee of Defence, but there is the old saying about doctors differing. Apart from this the Committee of Defence is not by any means, as is so often assumed, a body composed of politicians. On the contrary, there is included the First Lord of the Admiralty, the First Sea Lord and the Director of Naval Intelligence, while the Committee have the power to co-opt anyone as a member. Moreover, they will have at their disposal the fullest possible information from the Admiralty, and no one knows better the facts of war experience. It can be assumed that the naval staff have already analyzed with this full knowledge all that can throw light upon the selection of the most desirable warships for the future. Everyone with views backed by experience will be welcomed before the Committee, and when a decision is arrived at as to the type or types of craft desirable for fulfilling the vital work of Empire defence, it will be for the scientist, the naval architect, the engineer and other experts to embody in such ships the best that technical knowledge can devise. The one thing essential is that there shall be no heroics. Safety must come first.—"Engineering," Dec. 17, 1920.

NAVAL CONSTRUCTION IN 1920.

GREAT BRITAIN.

So far as this country was concerned, naval shipbuilding during the past twelve months remained virtually at a standstill. As in the preceding year, no war vessel of any description was laid down for the Royal Navy. Work was confined to the launch and completion of vessels left over from the war programme, and, with the exception of the light cruiser *Effingham*, there are no further British men-of-war on the stocks. The pause in British naval construction has been of longer duration than is realized by the world at large. Our two most modern battleships, the *Royal Sovereign* and *Royal Oak*, were laid down as far back as January, 1914. The battle-cruisers *Renown* and *Repulse* were begun exactly a year later, and the keel of the *Hood* was laid in September, 1916. A period of no less than fifty-one months has therefore elapsed since the last armored ship was laid down for the British Navy. Owing to this prolonged inactivity, which has coincided with unusual animation in the dockyards and arsenals of the United States and Japan, our relative position *vis-à-vis* those two Powers has sensibly declined. Recognition of this fact led the Admiralty, in the closing months of last year, to draft a new programme of construction, including, it is said, four capital ships of an improved type. In the meantime, however, doubts were raised in certain quarters as to the wisdom of spending money on the perpetuation of a type whose value in future naval warfare had been seriously questioned. Correspondence in the Press revealed a wide divergence of opinion on this subject. The Admiralty's own view had been clearly stated in a Memorandum published in March, 1920, from which the following passages may aptly be quoted:—

"There has been some criticism of the maintenance in commission of the present types of vessels, especially in regard to the capital ship. A contrary policy has been openly advocated, this policy being based, it is presumed, on the idea that the battleship is dead and that submersible and air vessels are the types of the future. The Naval Staff has examined this question with extreme care, and as a result we profoundly dissent from these views. In our opinion the capital ship remains the unit on which sea power is built up. So far from the late war having shown that the capital ship is doomed, it has, on the contrary, proved the necessity for that type. * * * On the British side the enemy submarines in no way interfered with the movements of capital ships in carrying out operations; destroyer screens, new methods of attack, and altered tactical movements defeated the submarine. Nor at the present could the Board of Admiralty subscribe to the statement that aircraft have doomed the capital ship. Aircraft are certainly of the highest importance in naval tactics, as regards reconnaissance, torpedo attacks, and artillery observation; but their *rôle* in present circumstances is that of an auxiliary and not of a substitute for the capital ship. The past history of this question must be taken into account; many times has the doom of the battleship been pronounced. The introduction of torpedo craft was believed in certain quarters over twenty years ago to have settled its fate. The Board of Admiralty at the time refused to be carried away by the attractiveness of the idea of building small, cheap torpedo craft instead of battleships, and they proved to be right. History has shown that the introduction of a type to destroy the capital ship has been quickly followed by the evolution of counter-measures which sustain its power. We therefore believe that the battleship must remain the principal unit, and that fleet tactics and tactical training must be carried out with the battle squadron as the main unit. Nevertheless, it must be emphasized that although the battleship remains, its type may require to be altered. Advances in electricity, in the internal combustion engine, and in science generally will inevitably necessitate an eventual change of type, and it is one of the principal functions of the Naval Staff to keep continuous watch on scientific development, with the object of ensuring that the type of capital ship designed meets the requirements of the future. It is even possible that the present battleship will change to one of a semi-submersible type, or even of a flying type, but such types are visions of the far future, not practical propositions of the moment. By gradual evolution and development the types forecast may arrive, but the immediate abandonment of the capital ship in favor of a visionary scheme of aircraft and submarines would leave the British nation destitute of sea power and without the means of progressive training."

The case against the capital ship has been stated with equal conviction by a number of naval officers, including Admiral Sir Percy Scott, Rear-Admiral Sydney S. Hall—who commanded the British submarine service for the greater part of the war—and Admiral W. H. Henderson. The last-named officer declares, in a letter to "The Times," that the principles which up to now have governed the capital ship no longer apply. They reached their maximum in the middle of the last century, when the big ship enjoyed a freedom of movement limited only by the weather, and a radius of action limited only by her three months' supply of fresh water; when the whole of her displacement, excepting only the weight of hull, stores, etc., and the comparatively small proportion required for sail propulsion, was devoted to offensive power. Since then, owing to the introduction of steam and armor and the gradual development of her antagonists, the torpedo, the submarine, the mine, the bomb, and the aerial torpedo—the powers of which will greatly increase in the future—the big

ship, Admiral Henderson contends, has lost her mobility, her freedom of movement, her radius of action, and her comparatively great offensive power. She is no longer supreme on the water, for when at sea her main object is to protect herself, and she dare not move without an escort of defensive auxiliaries. Protective devices are pre-empting an ever-growing proportion of her displacement, and though her speed—which is as much a defensive as an offensive factor—has been increased, her cost has become prohibitive. On the other hand, an equally strong case for the retention of the capital ship has been made out by various naval officers of distinction, including Admiral Sir Reginald Bacon.

In view of this conflict of professional opinion, the Government wisely decided to institute a close investigation into the whole question of naval strength as affected by the latest developments of naval warfare, before submitting any new programme of construction to Parliament. This inquiry, which is welcomed by the Admiralty, has been entrusted to the Committee of Imperial Defence, and the Prime Minister has given an assurance that every effort will be made to obtain the most competent evidence. In naval circles it is regarded as a foregone conclusion that a policy of compromise will eventually be adopted, provision being made, on the one hand, for the building of a limited number of capital ships, and, on the other, for research and experiment in connection with the development of submarines, aircraft, and other special types. The "improved type" of capital ship recommended by the Admiralty is understood to have been based on the *Queen Elizabeth* design, but with important modifications in regard to protection and armament. It goes without saying that the cost of building a ship to equal, let alone surpass, in fighting power the latest American battleships—43,200 tons displacement with twelve 16-inch 50-caliber guns—would be enormous. Unofficially, it is estimated at £9,000,000.

The battle-cruiser *Hood*, built by John Brown and Co., Limited, and launched on August 22, 1918, was commissioned on January 7, 1920, by Captain Wilfred Tompkinson, as flagship of Rear-Admiral Sir Roger J. B. Keyes, commanding the Battle-Cruiser Squadron, and after carrying out trials in the North Sea and Atlantic was completed to full complement on March 29, 1920. Full descriptions of this noble vessel have already appeared in our columns, but a few additional points of interest, brought out during the discussion at the Institution of Naval Architects in March last year, may be referred to. Commander E. S. Land, U. S. N., suggested that the *Hood* could do with more deck armor and less side armor, especially the 7-inch and 5-inch plates. He would have liked also to see thicker armor on the turret roofs. Generally speaking, however, he considered the ship to be an extremely well-protected unit. The design and arrangement of the conning-tower was one of the features most admired by American naval officers. He added that, while the American system of under-water protection was quite different, American constructors had undoubtedly obtained some excellent ideas from the design of the *Hood* and would no doubt use them. Sir Philip Watts drew attention to the great saving of weight that resulted from the adoption of geared turbines and small-tube boilers. Had these been available when the *Tiger* and *Queen Elizabeth* were designed, it would have been possible to give those vessels speeds of 32 and 28½ knots, respectively. By putting geared turbines and small-tube oil-fired boilers into a *Tiger*, lengthening her somewhat, and increasing her displacement by about 4,500 tons, she could be given the same protection against gunfire, the same armament, and the same speed as the *Hood*. Her displacement would then be about 33,000 tons, or about 8,000 tons less than that of the *Hood*. She

would, however, lack the bulge protection against torpedo attack. Rear-Admiral Sir A. Chatfield, Assistant Chief of Naval Staff, pointed out that although the *Hood* was to some extent a war production, she was not the result of *all* their war experience. The ship was originally designed before the Battle of Jutland, and was subsequently modified on the experience derived from that action. "At the present moment," concluded Admiral Chatfield, "we are full of experience; we are full of war lessons; we are full of new scientific ideas which have cropped up during the four years of war, to which we have not yet had time to give expression. We are full of knowledge and we want to use that knowledge in our new ships. We know that we can do that only by carefully studying all those lessons and applying them to our new designs. There is not the least doubt that if the Director of Naval Construction was going to design a new ship now he would not design the *Hood*, because, as I said, he designed that ship four years ago." Sir E. Tennyson d'Eyncourt, in replying to comments on his paper, mentioned that the total weight of the main propelling engines and boilers of the *Hood*, with 150,000 horsepower, was almost exactly the same as that of the *Renown* and *Repulse*, with the large-tube boilers, of which the designed power was 112,000, the figure actually attained being about 120,000. For the same weight, therefore, about 30 per cent more power was gained. In fact, had the small-tube boiler been fitted in the *Renown* and *Repulse*, it would very nearly have been possible, so far as weight went, to have added another turret with a pair of 15-inch guns to each of those ships.

The trials of the *Hood*—see a supplement this week—were entirely successful. On the measured mile off Arran she obtained a mean speed of 32.07 knots with 150,000 shaft horsepower; on a subsequent day, with about the same power and with a full load of fuel, the mean speed realized was 31.9 knots with all weights on board and at the deep load draught. These were very satisfactory results, for the speed of the ship with increased protection proved equal to the original designed speed of the more lightly protected vessel. Engines and boilers worked faultlessly throughout the trials, and economic oil consumption was obtained.

The most important naval launch of the year was that of H. M. S. *Frobisher*, which took place on March 20 at Devonport, where the vessel had been on the stocks for nearly three years. This vessel belongs to the "Improved *Birmingham*" class, consisting of four units. The *Hawkins*, which was the first to be completed, has been flagship in China since August, 1919. The *Raleigh*—see supplement—built by Beardmores, was launched on August 29, 1919, and handed over to the Admiralty on September 11 last, having completed some very successful trials. She is now about to leave for Bermuda as flagship on the North America station. The *Effingham*, the fourth unit of this class, was laid down at Portsmouth on April 2, 1917, as part of the war programme, without definite date for completion, but to be accelerated as required, Sir James Craig announced in Parliament last July that the vessel was then about 35 per cent complete so far as the dockyard work was concerned. The machinery, however, which has been built by contract, was then some 97 per cent complete, and the gun mountings were ready. It was expected that the slip occupied by the *Effingham* would be vacant about May, 1921. We may add that a personal visit to Portsmouth last August showed this vessel to be well advanced, and if the rate of construction has been maintained she should be ready for launching early this year. Although these four ships have been adversely criticized on account of their large dimensions and high cost, there can be no doubt as to their war value and general usefulness. They are at present the largest, fastest, most adequately

protected, and heavily armed cruisers afloat, and it is a remarkable tribute to their design that both the American and French naval authorities should be considering its adoption with certain modifications. We published a full description of the class in our issue of October 15 last.

The *Emerald*, a light cruiser of smaller dimensions, was launched at the Walker yard of Sir W. G. Armstrong, Whitworth and Co., on May 19, a sister ship, *Enterprise*, having been put afloat by John Brown and Co. in the previous December. A third vessel of this class, *Euphrates*, building by the Fairfield Company, was cancelled after the Armistice. The *Enterprise* and *Emerald* were designed in the winter of 1917-18 and laid down respectively in June and September, 1918. Being intended for ocean service, they have the substantial displacement of 7,550 tons, which therefore makes them intermediate between the "D" class, of 4,650 tons, and the *Raleighs*, of 9,750 tons. Other details are:—Length between perpendiculars 565 feet; beam, 54½ feet; depth forward, 30 feet 7 inches; amidships, 14½ feet; aft, 17 feet; deep load draught, 18½ feet. These two ships are designed for the unusually high speeds of 33 knots at light draught and 32 knots when fully loaded, and will thus be the fastest cruisers ever built for the Royal Navy. The propelling machinery of the *Emerald* has been manufactured by the Wallsend Slipway and Engineering Company, and that of the *Enterprise* by the builders, John Brown and Co. It consists of a double set of geared turbines operating four shafts, the turbines for the wing shafts being situated in the forward engine-room and those for the inner shafts in the after engine-room; each set comprises a high-pressure and low-pressure ahead turbine, with an astern turbine incorporated in the exhaust casing of the low-pressure ahead turbine. There are eight small-tube oil-fired boilers arranged in four compartments, three of which are before the midship magazines, and the fourth abaft the forward engine-room. The boilers in the second and third compartments are so disposed that the uptakes of all four are led into one large funnel. As separate funnels are provided for the boilers in the first and fourth compartments, the three funnels will be unequally spaced giving the ships an appearance not unlike that of the *Lion* class of battle-cruisers. The estimated power of the machinery is 80,000 shaft horsepower, a figure unprecedented in ships of this type, and hitherto exceeded only by the latest battle-cruisers. The maximum capacity of oil fuel will be 1,600 tons and the normal supply 650 tons. The *Enterprise* and *Emerald* have no bulge protection, but special attention has been paid to subdivision in and in way of boiler, machinery, and magazine spaces. The armament consists of seven 6-inch 50-caliber guns, five on the center line and two on the beam, giving a broadside of six guns; two or three 3-inch anti-aircraft guns, and twelve 21-inch torpedo tubes on four triple deck mountings. In view of the enormous power to be developed on such a moderate displacement, the steam trials of these two cruisers will be watched with keen interest. The *Enterprise* is completing at Devonport and the *Emerald* at Chatham Dockyard.

The two remaining flotilla leaders of the war programme, *Keppel* and *Rooke*, both built by John I. Thornycroft and Co., were launched at Southampton on April 23 and September 16, respectively. These ships are uniform with the *Shakespeare*, *Spencer*, and *Wallace*, built during the war by the same firm. Dimensions:—Length between perpendiculars, 318¼ feet; length over all, 329 feet; beam, 31¾ feet; depth, 19¾ feet. extreme draught, 14¾ feet; displacement, 1,740 tons. The propelling machinery consists of Brown-Curtis geared turbines and four Thornycroft

oil-burning boilers, to give a guaranteed speed of 36 knots at 40,000 shaft horsepower. The armament comprises five 4.7-inch guns, one 3-inch anti-aircraft gun, and six 21-inch torpedo tubes on two triple mountings.

A few belated war programme destroyers, mainly of the *Admiralty V* class, and several submarines have been completed during the year, among the latter being the M 3, a "submersible monitor" armed with a 12-inch gun. The L 33 and further units of the "L" class have also been delivered by various builders. Vessels of these two classes are illustrated in a supplement today. Experience has shown the "L" class of submarines to be the most efficient and generally useful for all-round service. They are of moderate displacement, about 1,070 tons submerged, have a length over all of 231 feet, and are propelled by a twin set of twelve-cylinder solid-injection motors developing 2,400 brake horsepower, the maximum surface speed being 17½ knots. For submerged running there is the usual plant of electric motors and storage batteries, giving a speed of 10.5 knots. In this class the problem of ventilation has been carefully studied. Air trunks are led through every compartment, and electric fans in connection with them keep the air as fresh as possible in the dynamo, engine, and torpedo-rooms and living compartments. The superstructure is much larger than that of earlier submarines, and is provided with emplacements for the two guns—4-inch and 3-inch—which thus have a high command, enabling them to be fought in weather that would make it impossible to work a gun mounted on the forward deck. The earlier group of "L" boats have four bow and two beam torpedo tubes; but in the later boats all six tubes are mounted in the bow.

Among the smaller vessels completed for the Navy in the past year were one or two coastal motor boats, built by John I. Thornycroft and Co. In the course of the official trials held in December a mean speed of 37 knots was achieved. During the war, it will be recalled, three types of C. M. B. were designed, the first model having a length of 40 feet and being armed with one torpedo. Two larger models were subsequently built, 55 feet and 70 feet long, respectively, and the armament was increased to two torpedoes, besides machine guns. It has been found, however, that the 55-foot type gives the best results, and the latest C. M. B.'s are accordingly of that model.

UNITED STATES.

At the end of October—the latest date for which official figures are available—the following war vessels and auxiliaries were under construction in the United States:—10 battleships, 4 battle-cruisers, 10 scout cruisers, 55 destroyers, 49 submarines, 2 gunboats, 2 fuel ships, 2 ammunition ships, 2 destroyer tenders, 1 submarine tender, 1 repair ship, and 1 hospital ship. Preparations were in train for the laying down of a further battleship and 2 battle-cruisers. During the twelve months ending October 1, 1920, there were completed:—1 battleship—*Tennessee*—84 destroyers, 12 submarines, 8 *Eagle* boats, 1 fuel ship, 1 ammunition ship, 1 gunboat, 6 mine sweepers and 11 fleet tugs. By the end of the year the battleship *California* had been delivered to the Navy. Latest reports indicate that the new battleship *Massachusetts* will be laid down in the first half of January. The scope of the American shipbuilding programme now in hand may be seen from the subjoined table:—

Battleships.

	Displacement. Tons.	Laid down.	Approximate date of completion.	Where building.
Colorado	32,600	1919	1922	Camden, N. J.
Maryland	32,600	1917	1921	Newport News
Washington	32,600	1919	1922	Camden, N. J.
West Virginia	32,600	1919	1922	Newport News
South Dakota	43,200	1919	1923	New York
Indiana	43,200	1920	1923	New York
Montana	43,200	1920	1923	Mare Island, Cal.
North Carolina	43,200	1919	1922	Norfolk, Va.
Iowa	43,200	1920	1923	Newport News
Massachusetts	43,200	1921	1923	Fore River

*Battle-cruisers.**

Lexington	43,500	1920	1923	Fore River
Constellation	43,500	1920	1923	Newport News
Saratoga	43,500	1920	1924	Camden, N. J.
Ranger	43,500	1920	1923	Newport News
Constitution	43,500	1920	1923	Philadelphia
United States	43,500	1920	1923	Philadelphia

Light Cruisers.

Omaha	7,500	1918	1921	Tacoma, Wash.
Milwaukee	7,500	1918	1921	Tacoma, Wash.
Cincinnati	7,500	1919	1922	Tacoma, Wash.
Raleigh	7,500	1920	1922	Quincy, Mass.
Detroit	7,500	1920	1922	Quincy, Mass.
Richmond	7,500	1919	1921	Philadelphia
Concord	7,500	1919	1921	Philadelphia
Trenton	7,500	1920	1922	Philadelphia
Marblehead	7,500	1919	1922	Philadelphia
Memphis	7,500	1919	1922	Philadelphia

Only two of the vessels listed above had been launched by the end of the year, viz., the battleship *Maryland* on March 20, and the scout cruiser *Omaha* on December 15. Delays have arisen owing to modifications in design, lack of building facilities—especially at the Government yards—and other causes, and it is possible that the entire programme will not be completed before 1924. As regards smaller vessels, it was stated in the House of Commons on December 15 that the United States had sixty-three destroyers and fifty-three submarines in hand, but these totals are somewhat in excess of the official American figures. As we propose to give, in an early issue, a detailed description of the larger vessels now building for the United States Navy, it is unnecessary to deal with them here. It may be mentioned, however, that in the case of the light cruisers the original displacement of 7,100 tons has been increased to 7,500 tons, and the original speed of 35 knots reduced to 33.7 knots, these modifications being due to the addition of four 6-inch guns on twin mountings at bow and stern, thus bringing the main armament up to twelve 6-inch 53-caliber quick-firing guns.

* The keels of three of these battle-cruisers were laid on the same day, viz., September 25, 1920.

JAPAN.

Next to the United States, Japan showed the greatest activity in naval construction during the past year. We are indebted to the Japanese Naval Attaché in London, Captain S. Kobayashi, I. J. N., for the following list of warships launched recently:—

	Launched.
Battleship, *Nagato*	November 9, 1919
Battleship, *Mutsu*	May 31, 1920
Light cruiser, *Kuma*	July 9, 1919
Light cruiser, *Tama*	February 10, 1920
Light cruiser, *Kitakami*	July, 1920
Light cruiser, *Ohi*	July, 1920

The *Nagato* and *Mutsu* are sister battleships of 33,800 tons displacement, with a designed speed of 23 knots. The main armament consists of eight 16-inch 45-caliber guns, with an auxiliary battery of twenty 5.5-inch quick-firers. They are reported, unofficially, to be 709 feet long over all, with a beam of 98 feet, and to have 13-inch armor on the water line. The horizontal protection is very good, having been designed with a view to offering strong resistance to plunging fire. These ships are said to be fitted with a "modified form" of bulge protection, "which does not interfere with speed." The four light cruisers named are of uniform design: 5,500 tons, 33 knots, seven 5.5-inch guns. The *Kiso*, a fifth ship of the same class, was laid down early in 1920 and is due to be launched in the coming spring. Two further battleships were laid down during the year, the *Tosa* at Nagasaki in February, and the *Kaga* at Kobe in July. These ships may be regarded as a "reply" to the American "*Indiana*" class to which they correspond in displacement and armament, but no official information is available. Captain Kobayashi informs us that four battle-cruisers, *Amagi*, *Akagi*, *Atago*, and *Takao*, have been or are about to be laid down. American reports credit these ships with a displacement of 42,000 tons, a speed of 34 knots, and an armament of eight 16-inch guns, in which case they would closely resemble the American battle-cruisers. The contracts have been placed as follows:—*Amagi*, Yokosuka Navy Yard; *Akagi*, Kure Navy Yard; *Atago*, Kawasaki Company, Kobe; *Takao*, Mitsu Bishi Company, Nagasaki. Four new light cruisers were also begun during 1920, viz., *Nagaro*, *Isudzu*, *Natori*, and *Yura*. They are to be sister ships, displacing between 6,000 and 7,000 tons, and designed for a speed of 34 knots. It is understood that the armament, which has not yet been determined, may include a few guns of larger caliber than 6-inch. Several large destroyers of 1,350 tons and 34 knots were launched in the course of the year, and eight or twelve new boats of a similar type were put in hand. Although no details of submarine construction are officially published, it is known that at least twenty of these vessels are now building for the Japanese Navy, the majority of which are of a large ocean-going type. In addition, seven ex-German submarines have been allocated to Japan for experimental purposes, the terms of the Peace Treaty requiring them to be broken up eventually. In July last a far-reaching scheme of naval expansion was presented to the Japanese Diet. This measure calls for the building of four battleships, four battle-cruisers, twelve light cruisers, thirty-seven destroyers, seventy-five submarines, and eighteen auxiliary ships, all of which are to be completed by 1928, at a total cost of 680,000,000 yen—£68,000,000 at par. The Minister of Marine, Admiral Kato, said that the programme was not aimed against any potential enemy, but was dictated by the country's insular position, "although the possibility of the dispatch of foreign strength to the Orient had not

been disregarded." He added that the building capacity of Japan was two capital ships a year—a very conservative estimate in view of the large extensions made since 1914 in the State-owned and private shipyards and armament factories. A slump in merchant shipbuilding set in twelve months ago, but the principal establishments such as the Mitsu Bishi Company, the Kawasaki Company, the Osaka Ironworks, the Yokohama Dockyard Company, the Uraga Dock Company, and the Asano Company, are occupied with important naval contracts that will keep them busy for the next two or three years.

FRANCE.

The French Government has not yet settled its future naval policy, and there has been in consequence very little activity in the dockyards since the war. The programme of 1919 included six cruisers, of 4,750 tons and 30 knots, with light armor protection and a battery of eight 5.5-inch guns; together with six destroyers of 2,000 tons and 35 knots, armed with three 5.5-inch guns. None of these vessels has been laid down yet, and the plans are likely to be modified. The Conseil Supérieur is now said to be considering the following designs:—

Capital Ships.

(1) 32,000 tons, 24 knots, six 17.7-inch guns.
(2) 35,000 tons, 28.5 knots, six 17.7-inch guns.
(3) 30,000 tons, 23.5 knots, eight 15.8-inch guns.

Light Cruisers.

(1) 5,300 tons, 32 knots, eight 5.5-inch 55-caliber guns.
(2) 6,000 tons, 35 knots, eight 5.5-inch 60-caliber guns.
(3) 7,600 tons, 33 knots, six 7.6-inch guns.
(4) 8,500 tons, 33 knots, five 8-inch guns.

Ten large submersibles are reported to have been ordered in 1920, but no details are forthcoming. Work in the French dockyards is therefore limited for the time being to the completion of several "avisos," or sloops, built under the war programmes, and to the reconditioning of the following ex-German war vessels alloted to France by the Inter-Allied Naval Council:—

Light Cruisers.

	New Name.
Regensburg	*Strasbourg*
Koenigsberg	*Metz*
Stralsund	*Mulhouse*
Kolberg	*Colmar*
Novara (ex-Austrian)	*Thionville*

Ten ex-enemy destroyers have also been incorporated in the French Navy, viz., S. 113 (re-named *L'Amiral Sénès*), H. 147 (*Delage*), H. 146 (*Rageot-de-la-Touche*), S. 134 (*Vesco*), S. 139 (*Deligny*), S. 133 (*Chastang*), S. 135 or 136 (*Mazaré*), V. 130 (*Buino*), V. 79 (*Pierre Durand*), and Ducla (*Matelôt-Leblanc*). Ten former German submarines have been placed under the French flag.

ITALY.

The Italian Government announced last summer that its plans for the near future did not include the construction of large ships of war. The defence of the coast would be entrusted to light mobile forces, and with the exception of torpedo craft and submarines all naval work had been countermanded. The battleship *Caracciolo*, 30,900 tons, laid down at Castellamare late in 1914, on which work had been suspended during the war, was launched in order to clear the slip, and will be completed as an oil tanker or a transport. Four flotilla leaders, 2,200 tons and 34 knots, armed with eight 4.7-inch guns, are building by Ansaldo and Co., at Genoa; but a group of six larger vessels of this type, ordered from Pattisons, of Naples, have been cancelled. The destroyer *Solferino*, 904 tons, 32 knots, four 4-inch guns, was launched by Orlando Brothers at Leghorn on April 28. Six small gunboats of a new type were laid down in the Pattison yard at Naples in May and June. Their names are *Andrea Bafile*, *Carlo del Greco*, *Tolosetto Farinati*, *Ernesto Giovannini*, *Emanuele Russo*, and *Allessandro Vetturi*. The displacement is 230 tons, speed 23 knots, and the armament consists of two 4-inch guns. It is uncertain whether these vessels will be propelled by steam or internal combustion engines. The Italian Navy has been reinforced by the addition of the ex-enemy light cruisers *Graudenz*, *Pillau*, *Strassburg*, *Helgoland*, *Saida*, and ten destroyers.

GERMANY.

On December 3 the German Reichstag approved a vote of 25,000,000 marks for the construction of a light cruiser. Socialist members, who opposed the grant, declared it to be nothing more or less than the inauguration of a new naval programme, and the new cruiser was to be "the manifestation of that old militarist spirit which was still alive." By the Treaty of Versailles Germany is inhibited from building light cruisers of a displacement greater than 6,000 tons. The new vessel will probably be laid down at Wilhelmshaven at an early date. The former Imperial Yard at Danzig has been engaged during the past year in converting several warships into merchantmen. The old cruiser *Gefion* is now the M. S. *Adolf Sommerfeld*, a cargo vessel fitted with two six-cylinder Diesel engines, each of 800 horsepower, which were originally manufactured for use in submarines. Another old cruiser, the *Viktoria Luise*, has been converted to the single-screw cargo steamer *Flora Sommerfeld*. The two light cruisers *Wiesbaden* and *Rostock*, which were in an advanced state of construction at the Armistice have been taken to Lübeck, where they will be reconstructed as cargo steamers.

SPAIN.

The light cruiser *Reina Victoria Eugenia* was launched on April 21 at the Ferrol yard of La Sociedad Española de Construcción Naval. This ship was ordered as long ago as 1914, but progress on her had been greatly delayed by the shortage of materials during the war. She is in essential features a replica of the British *Birmingham* class, as will be seen from the following particulars:—Length over all, 462 feet; extreme breadth, 50 feet. depth moulded, 25 feet, 7½ inches; mean draught, 15 feet, 9 inches; displacement, 5,500 tons. Nickel steel plating affords protection to the water line, in conjunction with a sloping deck. The machinery consists of Parsons turbines with a straight drive on two shafts, designed for 22,500 horsepower and a speed of 25.5 knots. There are twelve Yarrow boilers for coal and oil fuel. Both machinery and boilers

were made in the shops at Ferrol. The armament comprises nine 6-inch guns, four 3-pounder anti-aircraft guns, four Vickers automatic guns, and four 21-inch twin-mounted deck torpedo tubes. On the occasion of the launch and for services in connection with the Spanish Navy, King Alfonso conferred upon Mr. H. J. Spiers, the director of the works, the Grand Cross of Naval Merit. The Ferrol yard has already built three Dreadnought battleships for the Spanish Navy, and is now at work upon two new light cruisers of 4,650 tons and 29 knots. In November, however, it was reported that the company had decided to close its shipyard indefinitely, owing to the attitude of its workmen, who had refused to repair a ship.

MINOR NAVIES.

There is but little of interest to chronicle in connection with the navies of the smaller Powers. The Dutch Government, in a memorandum published last June, laid down the principle that the defence of Holland by sea should be undertaken with "small material," such as submarines, minelayers, aeroplanes, and guard vessels. But as the defence of the colonies necessitated a few fast light cruisers and destroyers, the Government had decided to continue the construction of the cruisers *Java* and *Sumatra*, building at Flushing and Amsterdam, respectively. These ships, which have been previously described in "The Engineer," displace 7,050 tons and have an estimated speed of 30 knots. They were to have been built under the technical supervision of the Krupp Germania Yard of Kiel, which was also to supply materials and a certain number of skilled workmen, but that this arrangement has been adhered to is doubtful. Three submarines of ocean-going type are building at Flushing and Rotterdam.

Rumors that were current early last year to the effect that large orders for naval construction were pending from Brazil and Chile have not yet materialized. On the other hand, Chile has purchased the British battleship *Canada*, which has been re-named *Almirante Latorre*, and the flotilla leaders *Broke*, *Botha*, and *Faulknor*. All four vessels were begun originally for Chile and taken over by the British Admiralty during the war.—"The Engineer," Jan. 7, 1921.

BOOK REVIEWS.

OIL FUEL BURNING IN MARINE PRACTICE. BY J. W. M. SOUTHERN. D. VAN NOSTRAND COMPANY, 8 WARREN STREET, NEW YORK. 200 PP. 102 ILL. $7.50 NET.

The author states that the volume has been compiled by request to meet the demand of marine engineer officers for a book containing clear and exhaustive descriptions of various systems of oil fuel burning, combined with general practical information and hints relating to the working of this modern method of steam raising.

He gives the advantages and disadvantages of oil as fuel, explains briefly the chemistry of combustion, gives definitions of the usual oil terms, specific gravity tables, explanations of the various tests, and descriptions of the instruments used in making such tests. An occasional problem with its solution, is also given. The necessity for fittings and accessories is explained and the main features that an oil fuel system should possess are listed.

The various pressure systems of oil fuel burning are then described in detail, with many large plates and numerous illustrations. The systems described are the following: Kermode, Wallsend-Hawden, White Low Pressure, Smith's, J. S. White, Babcock and Wilcox, and Dahl.

Having described the various systems, the author proceeds to point out faults frequently found in oil burning, giving both cause and effect, and follows this by a series of questions and answers on the use of oil fuel. The last section is a summary of general information on the subject of oil fuel and includes Lloyd's Rules for burning and carrying of oil fuel.

The volume is a good one for the practical marine engineer.

INDUSTRIAL OIL ENGINEERING. By JOHN ROME BATTLE. J. B. LIPPINCOTT COMPANY, PHILA. 1131 pp. ILL.

A handbook, rather larger than customary, containing sections on mathematics and useful data; marketing of oils; mechanical engineering; petroleum and its products, including short paragraphs on the various theories of its origin, and characteristics of the various crudes; description of certain methods of refining and also the manufacture of lubricating greases; fats and oils other than petroleum products; testing and properties of oils with descriptions of certain instruments used in making tests; lubrication and friction with a theory of lubrication; lubricating and industrial oil equipment such as filters, purifiers, and coolers; oil storage and handling; lubrication of artillery and ordnance, aviation engines, automobiles, cranes, engines, elevators, locomotives, textile and mining machinery; and Government specifications for lubricants and their testing.

The information contained is rather general and brief in some instances and much more in detail in others, but this could probably not be avoided in the wide range of the subjects covered. However, as a reference book it should be useful.

THE MARVELS OF THE SHIP. By E. KEBLE CHATTERTON, LT. COMDR., R. S. V. R. J. B. LIPPINCOTT COMPANY, PHILA. 1921. 259 pp. 15 ILL.

In this work is set forth in some detail the growth of the ship from the first vessels of which there is any record in history down to the modern liner and war vessel, with certain theories as to the methods of transportation on water before the existence of the first known boat. The volume contains descriptions of the first ships of the Nile and the Mediterranean, with certain details of their rig and construction, a history of the evolution and development of the ship in various

countries at different dates, the first steam-propelled vessels, and modern vessels. A chapter is also devoted to the fishing fleets. Although no mention is made of new types which have been developed in very recent years, the book is an interesting one.

ASSOCIATION NOTES.

The Annual Banquet of the Society, which had been suspended during the war, was resumed this year, and was held at the New Willard Hotel, Washington, D. C., on December 18, 1920.

One hundred and sixty members and guests attended and the affair was most enjoyable. Captain John Halligan, Jr., U. S. N., presided as toastmaster and during the evening the following responded to toasts:

The Navy, Its Relation to the Foreign Policy of the United States, Senator Key Pittman.

Naval Engineers, Their Mission as Viewed by Command Afloat, Admiral R. E. Coontz, U. S. N., Chief of Naval Operations.

The Shipbuilder, Mr. J. W. Powell, Vice President, Bethlehem Shipbuilding Corp.

The Development of Engineering Talent in the Navy, Mr. W. L. R. Emmet, General Electric Co.

The Trend of Naval Engineering During the World War, Rear Admiral R. S. Griffin, U. S. N., Chief of Bureau of Engineering.

Fuel Oil for the Navy, the Secretary of the Navy.

The addresses of the Secretary of the Navy and Mr. W. L. R. Emmet will be found in this number of the JOURNAL.

The dinner was in charge of the following committee:

Commander F. J. Cleary, U. S. N.
Commander H. A. Stuart, U. S. N.
Lieut. Commander F. S. Sherman, U. S. N.
Lieut. Commander W. D. Chandler, U. S. N.
Lieut. Commander J. Q. Walton, U. S. C. G.

ELECTION OF OFFICERS.

The meeting to count votes for officers of the Society for the year 1921 was held at the Navy Department on Thursday, December 30, 1920. The following were declared elected:

President:

Captain A. J. Hepburn, U. S. N.

Secretary-Treasurer:

Commander J. S. Evans, U. S. N.

Council:

Rear Admiral B. C. Bryan, U. S. N.
Captain Robert Stocker (C. C.), U. S. N.
Commander S. C. Hooper, U. S. N.
Commander Kenneth Whiting, U. S. N.
Lieut. Commander M. R. Daniels, U. S. C. G.

FINANCIAL STATEMENTS, 1920.

PUBLICATION STATEMENT.

The following statement shows the cost of producing the JOURNAL of the Society for 1920 and the returns therefrom, with the gross profit:

	Costs.	*Returns.*
Dues		$5,801.18
Subscriptions		3,146.65
Sales		769.96
Advertisements		2,612.50
Exchange		2.43
		$12,332.72
Printing	$7,386.81	
Engraving	1,109.20	
Drafting	18.00	
Manuscript	747.00	
Commission	197.65	
Postage and Express	181.62	
		9,640.28
Gross Profit		$2,692.44

GENERAL PROFIT AND LOSS STATEMENT.

The following statement shows the result of the complete operation of the business during the year 1920:

	Loss.	*Gain.*
Gross Profit from Publication		$2,692.44
Interest		618.63
		$3,311.07
General Expense	$1,953.31	
Banquet, 1920	190.15	
Current Profit and Loss	519.52	
		2,662.98
Net Gain		$648.09

BALANCE STATEMENT.

The following statement shows the difference between the assets of the Society on January 1, 1920, and on December 31, 1920, with the net gain:

ASSETS.

Cash	$3,984.07
Accounts Receivable, Advertisements	1,783.14
Accounts Receivable, Dues	996.25
Accounts Receivable, Sales	56.79
Accounts Receivable, Subscriptions	975.00
Army and Navy Club Bond	1,000.00
Certificate Deposit, Munsey Trust Co.	1,062.76
Furniture	110.00
Government Bond, 1st Liberty	1,000.00
Government Bond, 3d Liberty	1,000.00
Government Bond, 4th Liberty	2,385.20
Government Bond, 5th Liberty	1,000.00
War Savings Stamps	838.00
Washington Railway & Electric Co. Bonds	4,320.00
	$20,511.21

LIABILITY.

Camden Forge Company	$50.00
Net Assets December 31, 1920	$20,461.21
Net Assets January 1, 1920	19,813.12
Net Gain	$648.09

An auditing committee has been appointed and accounts are now being audited.

MEMBERSHIP.

The Society desires to extend its membership in the Navy and it is requested that all members bring to the attention of officers, and especially the younger officers, with whom they are associated the advantages of membership and extend to them a most cordial invitation to join with us.

The following members and associates have joined the Society since the publication of the last JOURNAL:

MEMBERS.

Brimmer, K. F., Ensign, U. S. N.
Hamilton, J. E., Ensign, U. S. N.
Huntington, C. M., Ensign, U. S. N.
Jones, John W., Ensign, U. S. N.
Kennedy, S. S., Lieut. Commander, U. S. N.
Kime, F. D., Ensign, U. S. N.
King, James L., Lieut. Commander, U. S. N.
Klein, H. S., Lieut. U. S. N.
Merkt, A. J., Lieut. U. S. N.
Myers, G. B., Ensign, U. S. N.
Schneider, N. F., Ensign, U. S. N.
Van Auken, F. T., Lieut. Commander, U. S. N.

ASSOCIATES.

Bailey, Wm. H., Bureau of Engineering, Navy Dept.

Coward, Herbert, 501 Washington Loan & Trust Bldg., Washington, D. C.

Crossley, Alfred, 5320 Illinois Ave., N.W., Washington, D. C.

Davis, Hugh, care Goulds Mfg. Co., 111 North 3d St., Philadelphia, Pa.

Florit, Felix, Engineer Lieut., Argentine Navy.

Ford, Sherman, 25 Primrose St., Chevy Chase, Md.

King, J. H., care Babcock & Wilcox Co., 85 Liberty St., New York City.

Mackenzie, K. G., The Texas Co., 17 Battery Place, New York City.

Murtaugh, Walter A., Bath Iron Works, 17 Battery Place, New York City.

Speck, Wm., Speck Mfg. Co., 255 Octavia St., San Francisco, Calif.

Watson, Wallis C., 1034 Stanford St., Schenectady, N. Y.

Webster, Fred F., Woolworth Building, New York City.

ADDRESSES.

The Secretary-Treasurer again earnestly solicits your support in his effort to ensure prompt delivery of your JOURNAL by prompt notification of all changes of address.

MEETING OF AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

The Spring meeting of the American Society of Mechanical Engineers will be held at the Congress Hotel, Chicago, Ill., May 23 through May 26, 1921. Sessions are planned on Aeronautics, Fuels, Management, Material Handling, Machine Shop, Power, Forest Products and Railroads.

Errata.

Vol. XXXIII, No. 1.

Table "SUMMARY OF DATA" immediately preceding page 11:

In the horizontal line giving "LBS./S.H.P./HR.—T.O.", under 15 knots read "18.0" for "17.3" and under 30 knots read "10.65" for "11.65".

Page 140, line 11; for:

$$\text{I. H. P.}_d \text{ (corrected)} = \text{I. H. P.}_{d^1} \times \frac{D_1}{D}$$

read $\text{I. H. P.}_d \text{ (corrected)} = \text{I. H. P.}_{d^1}$.

Line 16; for:

$$R_d \text{ (corrected)} = R_{d^1} \times \left(\frac{D_1}{D}\right)^{\frac{1}{3}}$$

read $R_d \text{ (corrected)} = R_{d^1} \times \left(\frac{D_1}{D}\right)$

Line 19; for:

$$\text{I. H. P.}_{d^1} = \text{engine power} \times \left(\frac{D}{D_1}\right)$$

read $\text{I. H. P.}_{d^1} = \text{engine power.}$